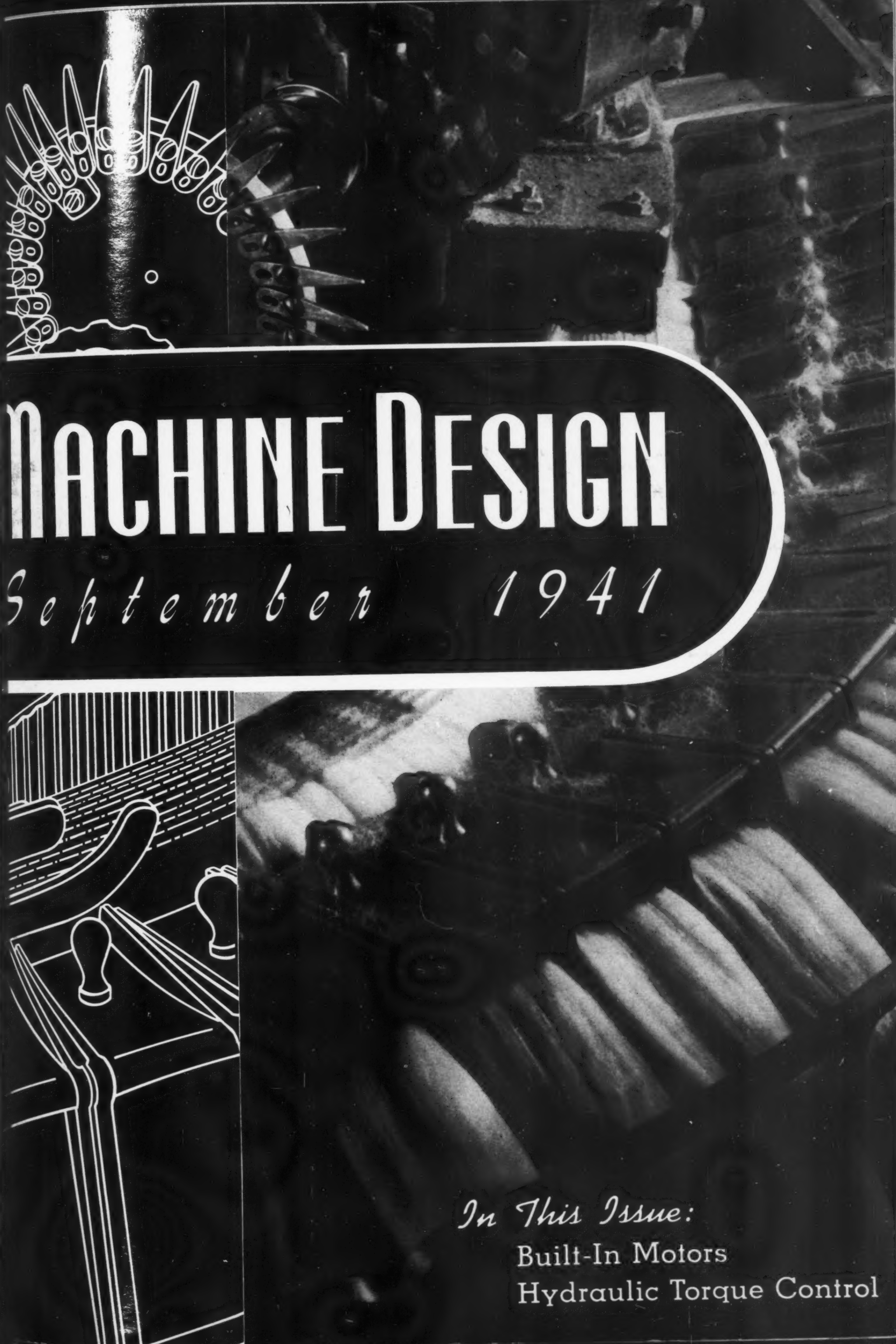

MACHINE DESIGN

September 1941

In This Issue:

Built-In Motors

Hydraulic Torque Control

...To Solve TUBING PROBLEMS ...To Safeguard UNIFORM QUALITY

...ONE OF THE LARGEST AND MOST COMPLETE CHEMICAL AND PHYSICAL LABORATORIES OF ITS KIND

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MECHANICAL TUBING

Topics

TO meet the present emergency, many machinery manufacturers are designing "stripped down" machines. These models ease somewhat the bottleneck experienced by defense industries. In addition they conserve needed materials otherwise used for refinements. An example is a munitions engraver built by the George Gorton Machine Co. that can be produced nearly three times as fast by eliminating many controls and adjustments used on standard machines.

♦

SCIENTIFIC conservation of materials is necessary for almost all machinery manufacturers. Careful reclamation and sorting of scrap becomes an important part of any program of this kind. As an indication of what a well organized effort can accomplish along this line, General Electric has developed its reclamation system to the point where each dollar's worth of materials saved is reclaimed at a cost of only twenty cents.

♦

DAY of perfection seems to be here for the homely typewriter ribbon. Latest development includes a ribbon that does not clog the type, does not smudge and has long wear with uniformity of color. Undetectable erasures can be made positively and quickly.

♦

ALUMINUM shortage troubling designers is being offset in various ways. In the radio industry, high-frequency iron is releasing considerable amounts of the metal as well as making marked effects on the trends in radio design. Iron-core coils, powder-iron sleeves, and permeability tuners eliminate the use of aluminum shields, condenser plates, etc.

One manufacturer has conserved more than 33 tons of aluminum by the use of steel in the manufacture of radio receiving set cases for the Army. These cases are copper plated before and after assembly. Then zinc chromate primes the cases preparatory to receiving regulation finish.

IN aircraft manufacture, aluminum rivets are kept in refrigerator compartments after heat treatment to keep them soft. Reaching normal temperatures after driving, the rivets attain their maximum strength and hardness.

♦

MACHINE tools manufactured in South America are now being imported as stop-gaps for domestic production. These include shapers, drill presses, punch presses, die filing machines and hand screw machines. Later imports will include large turret lathes, planers, heavy duty drill presses and milling machines built to conform with American specifications and practices.

♦

CELLULAR rubber, consisting of thin dense layers enclosing microscopic cells of nitrogen, has been developed which is half the weight of cork and has high insulation properties. Ten times more resistant to water than cork or balsa wood, the material will be furnished in two weights of insulation board by the United States Rubber Co. A thermoplastic material, it can be shaped by heating to 190 degrees, bending while hot. Other properties include machinability, structural strength, moisture and rot resistance as well as resistance to oil, acid and fire.

♦

COMPRESSED wood impregnated with phenolic resins is being used successfully for gears. They are relatively strong, quiet-operating and long-lived.

♦

BUTADIENE synthetic rubbers are being used as substitutes for spring steel, machine supports, shaft couplings, rotary oil pumps and cable coverings. Plasticized phenolic resins are also being used as substitutes, for coating and protecting tin plate and unplated iron.

Ingenious Mechanism Typifies Design of Jig Grinder

By J. R. Moore
Moore Special Tool Co.

INTERCHANGEABILITY of machine parts is one of the outstanding contributions of engineers to American progress. One of the best examples in point is that of the European skeptic who disassembled three airplane engines, shuffled the parts and reassembled the three engines in perfect working condition.

Such precision is the result of the accurate tools, dies and gages used to form and check the critical shape and dimensions of each machine part. Equally accurate, or even more so, must be the machines on which such tools and dies are made.

Grinding of cylindrical holes necessitates relative rotation of the wheel and the work about two axes—first, the axis of the hole and, second, the axis of the wheel. However, in addition to cylindrical holes, tapered holes must be ground. Furthermore, because the draft of holes for punching dies is opposite that of molding dies, the machine must be capable of grinding tapers in both directioins. Engineers familiar with the tedious use of toolmaker's buttons or locating plugs to achieve precise dimension and location of holes will appreciate the need for a machine fulfilling the above basic specifications for

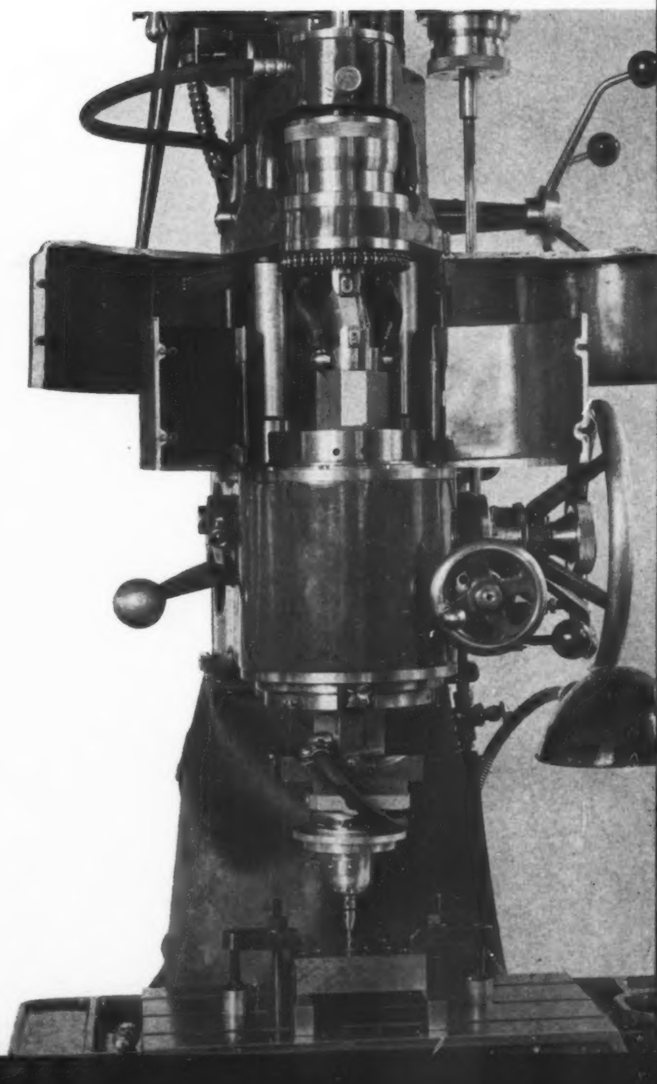
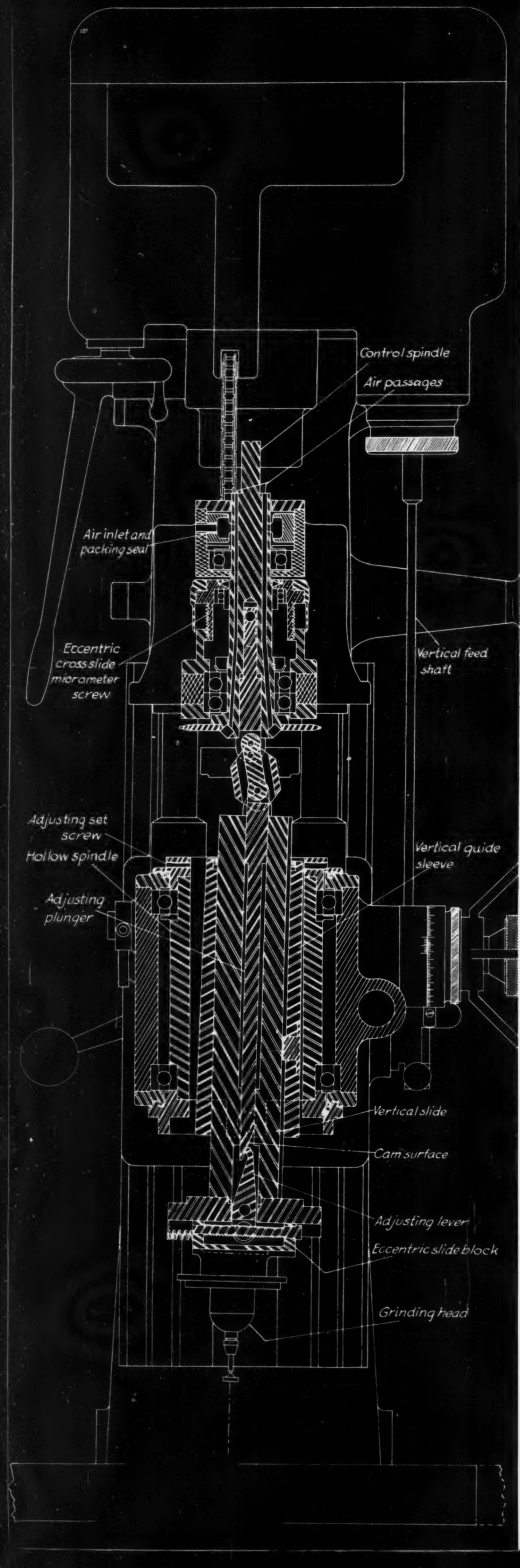


Fig. 1—Grinder finishes cylindrical holes and holes tapered in either direction to a high degree of precision



grinding holes in dies and other machine parts requiring high finish and dimensional accuracy.

Since die holes may vary from $\frac{1}{8}$-inch to 4 inches in diameter, wheel speeds from about 10,000 to 75,000 revolutions per minute are required. Also, in order to feed the wheel into the work, fine adjustments of the cutting diameter must be made while the machine is running.

It is evident from the foregoing that an ordinary internal grinder meets all the requirements except that of location. Therefore, the preliminary design visualized a vertical internal grinder with a rotating horizontal worktable the position of which could be moved in either of two directions with respect to its center of rotation.

In such a machine the grinding spindle would be mounted on a vertical slide which would be inclinable to permit taper hole grinding. The vertical slide could then be made adjustable horizontally to provide the "adjustable-while-running" feature.

As the design developed, certain apparently insurmountable difficulties were perceived. In order to handle a practical range of sizes of work the rotating table would necessarily be large, introducing high centrifugal forces. Due to large bearing sizes required and their heavy loading, extreme bearing problems were introduced. Also, heat generated in these bearings would cause warpage and distortion. Finally, the element of cost advantage in using a conventional jig borer cross-slide worktable added to the lack of feasibility of this design.

In consequence, the whole idea of the rotating table was discarded. Consideration was given to various mechanical movements which would fulfill all the requirements and which could be built entirely into a single vertical spindle housing.

Rotates about Two Axes

The final solution is the machine shown in the photograph, *Fig.* 1, and in the drawing, *Fig.* 2. Solving, as it does, the problem of grinding cylindrical and tapered holes and providing rotation about two mutually inclined axes, the spindle assembly can well be considered in some detail.

The first axis of rotation is always vertical and always coincides with the axis of the hole being ground. A hollow spindle mounted in the housing in a pair of

Fig. 2—Left—When the slide is tilted as shown, machine is set up to grind tapered holes.

Fig. 3 — Installed and interchangeable grinding wheel assemblies. These heads are removed and replaced quickly by means of pear-shaped flange holes and quick-acting hose couplings

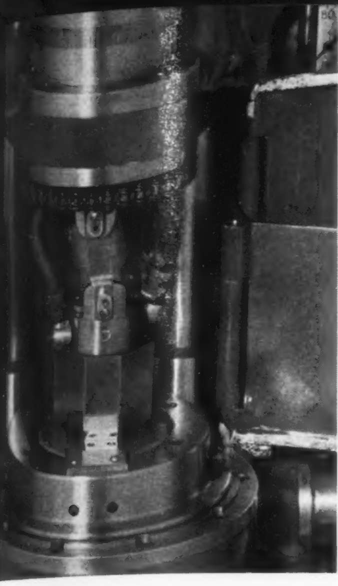
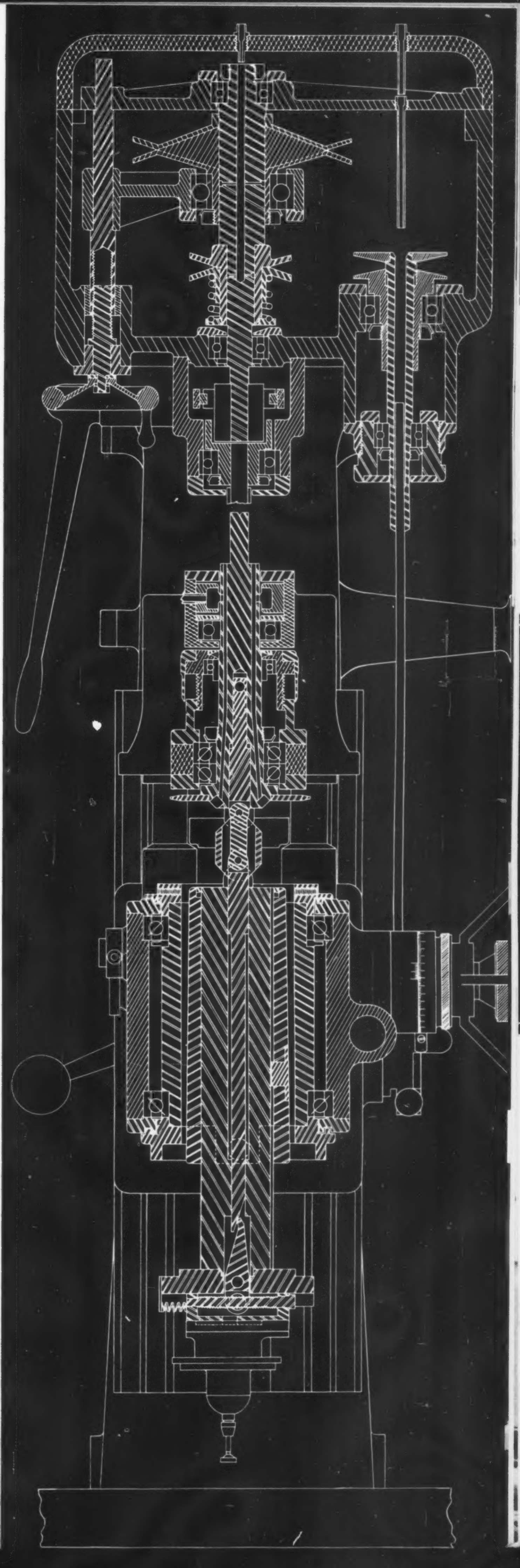


Fig. 4 — Left — Semi-universal link acts as a crank to drive spindle when slide is tilted. Chain provides spindle drive from jackshaft

Fig. 5 — Right — Drive, utilizing variable speed pulleys, is shown in head. Slide, adjusted to the vertical, grinds cylindrical holes



preloaded precision ball bearings determines this axis.

To provide the inclined axis for grinding tapers, a vertical guide sleeve is mounted inside the spindle. This sleeve is hinged on two diametrically opposed pivotpins integral with the lower part of the spindle. Since the pins are located in the front and back of the spindle they are not shown in the sectional view, *Fig.* 2. The upper end of the guide sleeve is clamped between two setscrews tapped into the spindle. By locating these screws in a line at a right angle to the center line of the pivotpins, the guide sleeve may be tilted in either direction by loosening one screw and taking up on the other. Any angle up to 1½-degrees in either direction may be set by means of calibrations on the spindle and sleeve.

Air Motors Drive Wheel

Support for the grinding wheel assembly is provided by a slide fitted into the hollow guide sleeve. The grinding wheel spindle drive (rotation of the wheel about its own axis) is conventional. Air motors are used, coupled to the slide by means of neoprene hose. Wheel speed variation is obtained by interchangeable motors. One of these is shown installed and another resting on the table in *Fig.* 3. To facilitate assembly of the wheel heads to the spindle cross slide, four pear-shaped holes are formed in the four corners of the square flange of the air motor housing. Four capscrews are tapped into the eccentric slide block. By loosening these screws and turning the air motor flange slightly, the capscrew heads are aligned with the large end of the pear-shaped holes and the entire assembly can be quickly removed and replaced.

When, as shown in *Fig.* 5, the slide is adjusted to the vertical by means of the setscrews in the upper edge of the hollow spindle, cylindrical holes are ground. By moving the grinding head laterally on the eccentric cross slide the diameters of the cylindrical holes are varied.

For grinding taper holes, the slide is tilted. The lateral adjustment of the grinding head on the cross slide then serves a double purpose. It will be recalled that holes must be ground tapered in either direction. If it is assumed that the center line of the grinding wheel coincides with the center line of the tilted vertical slide, a hole will be ground which tapers out-

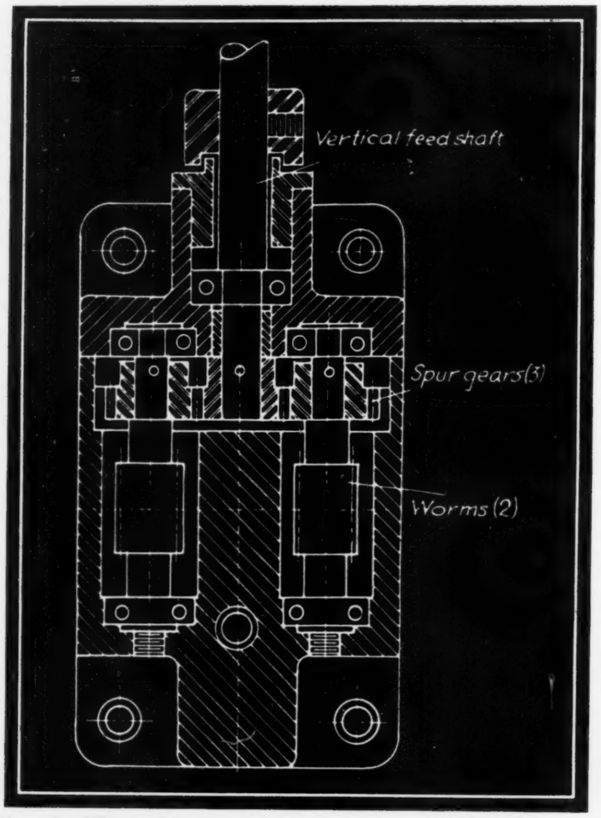
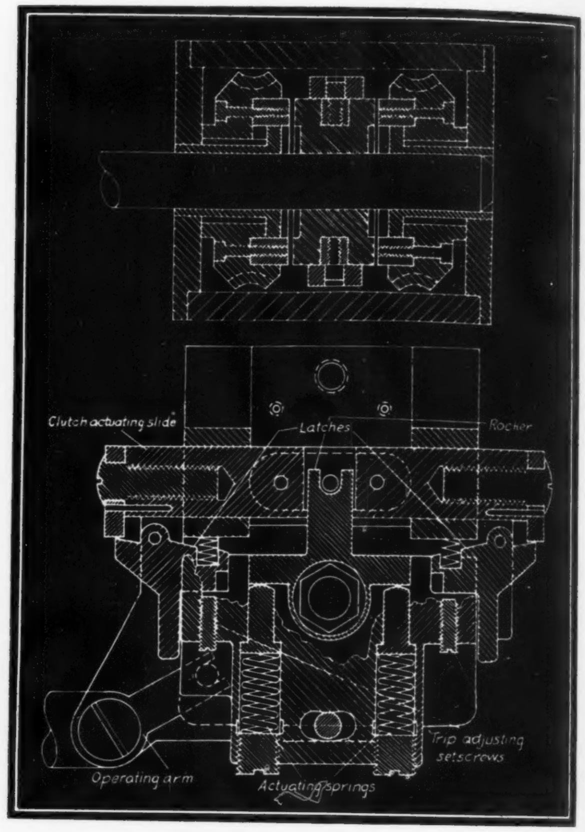


Fig. 6—Worms for vertical traverse are driven by spur gears engaged by pinion on vertical shaft

Fig. 7—Trigger mechanism shown in lower view affords precise control cf vertical traverse reversals



ward as the wheel is fed downward.

To grind a taper in the opposite direction it is apparent that the center of the wheel must be so located that, in being fed downward along the center line of the guide sleeve, it approaches the center line of the hollow spindle. This is accomplished by tilting the guide sleeve to the left in *Fig.* 2 and moving the grinding wheel to the left. Thus the mean diameter of the portion of the hole being ground by a cylindrical element of the wheel at any instant is equal to the total eccentricity of grinding element with respect to the guide sleeve less the length of a line perpendicular to the center line of the guide sleeve at the wheel center to its intersection with the center line of the spindle, multiplied by twice the cosine of the angle of tilt.

In contrast, the corresponding mean diameter of a hole which tapers outward as the wheel is fed downward is equal to the length of a line perpendicular to the guide sleeve center from the grinding element to its point of intersection with the spindle center line, multiplied by twice the cosine of tilt angle.

Rotation of the entire spindle assembly about its own vertical center was subject to a number of solutions. The simplest would be to mount a sprocket directly on the hollow spindle and drive from there. However, such a drive would have certain disadvantages.

First, the chain would impose a constant, unidirectional, radial load on the spindle which might detract from grinding accuracy. Secondly, the fine adjustment or displacement of the grinding wheel assembly on the eccentric cross slide must be made while the machine is running. Difficulties would be introduced in applying a stationary micrometer screw for this adjustment to the top of the slide which may be either vertical or inclined at various angles. A similar problem would be experienced in feeding the slide through the guide sleeve for traversing the wheel down the walls of the hole.

Universal Provided by Link

For these reasons, it was decided to use a separate control spindle. This shaft would be always vertical and concentric with the hollow spindle. Further, it would be capable of vertical axial movement. In order that this movement may be transmitted to the vertical guide even though this part is tilted it is connected to it by means of a link shown in *Fig.* 4 as well as in the drawing, *Fig.* 5. Demonstrated also by *Fig.* 4 is the manner in which the guide sleeve is tilted by the hollow setscrews as well as how the link, under this condition, acts as a crank to transmit rotation to the hollow spindle.

Chain drive of the control spindle is accomplished by driving the chain in *Fig.* 4 from a second sprocket mounted on a jackshaft running up the machine column. Variation in speed of rotation of the spindle

(*Continued on Page* 122)

Scanning the Field FOR IDEAS

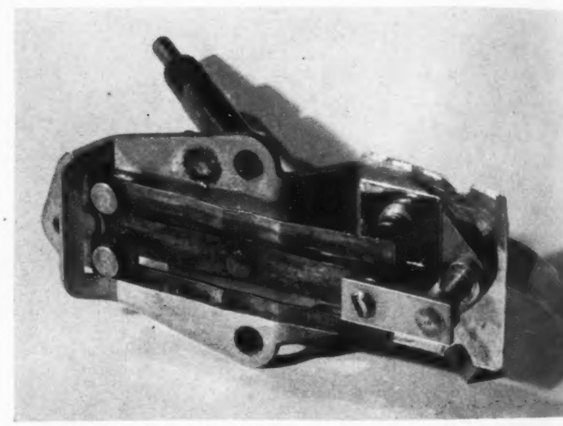
Traditional mechanisms for one type of machine have often furnished the solution to an otherwise difficult problem in another machine far afield from the original application. As an example, the latest RCA development for playing recordings on both sides was suggested by passenger train operation. Backing up for coupling cars indicated the idea of reversing the direction of rotation of a turntable to permit playing the underside of a record.

Shown in the photographs is this new development. Starting the cycle, the first record drops and plays on top in the usual manner. Tone arm then swings out, turntable reverses and the tone arm swings back to play from the bottom with a duplicate facing pickup. After playing, tone arm swings out again and turntable spindle withdraws from record which tilts and slides into a record compartment. Sequence then starts over and machine stops automatically after the last record is played and deposited.

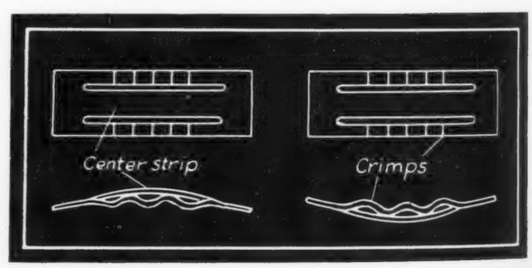
Two motors are utilized. Driving the turntable is a motor controlled by a reversing switch, pawl-operated from the cycle mechanism. Cycle motor operation is initiated by a mercury switch, the position of which is determined by the movement of the tone arm after finishing the record.

Bullet-proof hose, left, is an outgrowth of the self-sealing tanks for protecting vital fuel tanks in combat airplanes. Developed by B. F. Goodrich Co., the hose, in addition, provides substantial weight saving in the average bomber. In the test illustrated, a marksman punctured one hose five times with 50-caliber machine gun bullets yet the hose held fuel under 15-pounds pressure without leaking.

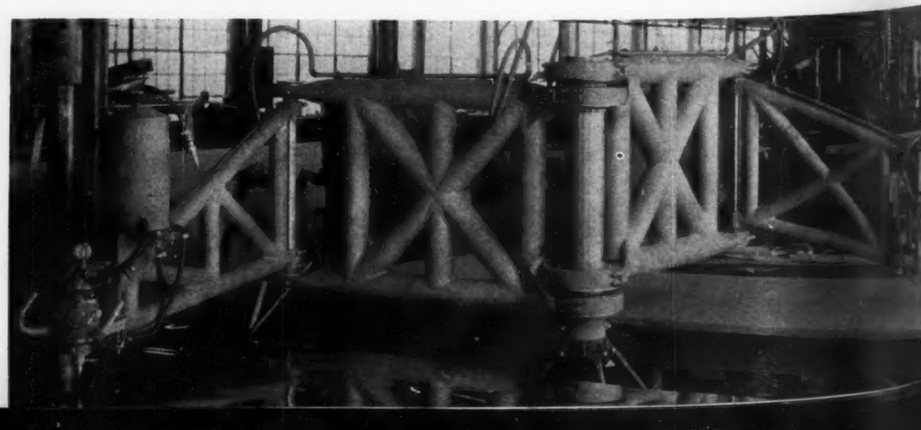
Hose has an unusually effective sealing member and is lined with Ameripol, a synthetic rubber resistant to oil and gasoline. Total wall thickness is less than 5/16-inch.

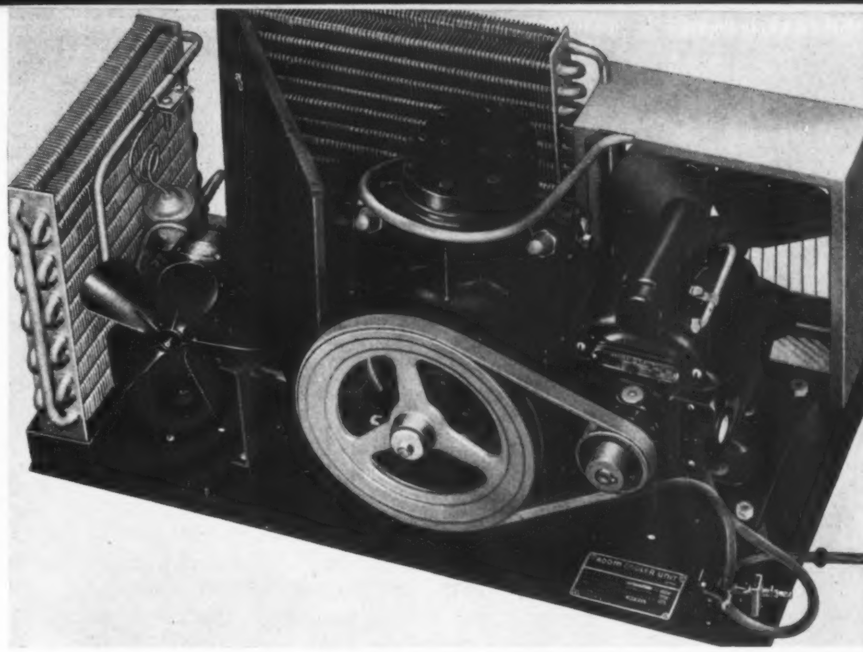


Crimp improves accuracy of the bimetal strip thermostat at right, making it suitable for operation within two degrees temperature change. Developed by Westinghouse, an element weighing 1/10-ounce closes with a force of five pounds. Made of strip stock and slotted to make three legs, the two outer legs are crimped by rolling between special gears, controlling the crimp within .0001-inch. This makes the outer legs shorter than the center which causes the strip to bow as shown in the drawing. Increase in temperature expands the bimetal and causes it to buckle in the opposite direction with a quick snap action. By controlling the thickness of the bimetal, the leg length and amount of crimp, it is possible to predetermine accurately the characteristics of a thermostat.



Aircraft welded-tubing construction is employed in the router, right, except that no particular attention is given to strength-weight ratios. Deriving operating efficiency from almost perfect balance of the supporting arms, the machine is built of heavy iron pipe without bends. Angle cuts made in the ends are similar to those used in aircraft fuselage construction. Ease of movement of arms is obtained with grease-packed ball bearings at the points where the structure is hinged together with yoke fittings. Central supporting post, requiring extreme rigidity, is based on concrete foundation and braced by four heavy arms extending pyramid fashion from the four corners of the base.



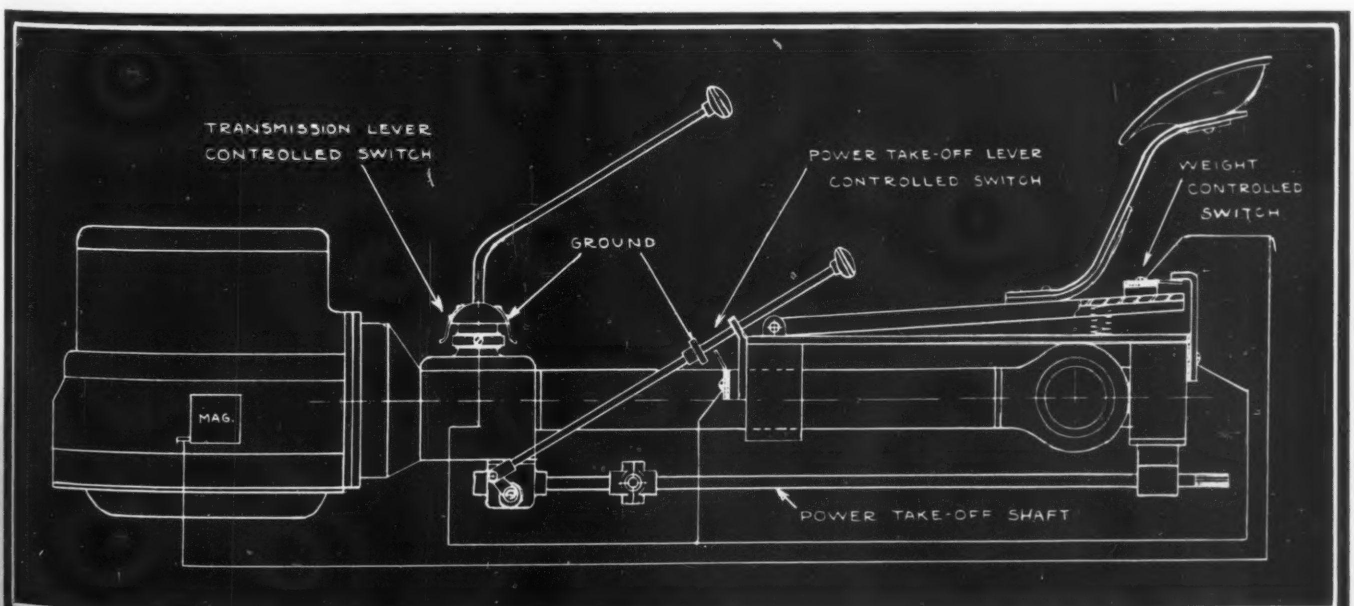


Quiet operation with minimum vibration is particularly necessary in domestic machines such as room coolers for operation throughout the night in bedrooms. In the Gale unit shown at right noise levels are reduced by improved mountings, baffles and balance of parts as well as appropriate operating speeds. Two evaporator coils turned at an angle on two sides of the chassis reduce restriction of air movement and attendant noise.

Compressor is carefully insulated by a floating base and is a high-speed, small bore unit. This type of design simplifies balancing problems in that the force of each power stroke is low. Pulsations, 1500 a minute, are required to be as smooth as possible. Smoothness is attained by casting a gas expansion chamber integral with the compressor body. An orifice is provided at the point where the compressed gas is again allowed to expand, making possible the use of smaller tubing for the discharge line and further aiding quiet operation. Condensed water is removed by atomizing and spraying it on exhaust collars.

Safety in operation is closely associated with simplicity of control and interlocking circuits. A noteworthy example of a simple, built-in safety feature is the "dead man" control developed by C. H. Dooley and illustrated in the schematic drawing of the tractor below. This control is arranged to ground the magneto during unsafe operating conditions, thus cutting off the spark to the engine.

As shown, a weight-controlled switch is in series with two parallel ground circuits—one to the transmission lever and the other to the power take-off lever. If either lever is in gear and the operator is not in control position, the magneto is grounded and the tractor is inoperable. Under this condition it is impossible to crank the engine while in gear or with power take-off connected. Also, if the operator should fall from his position, the tractor would stop immediately. In addition, should the machine upset, the spark would be disconnected, removing possible fire hazard.



Applying Theory of Elasticity

By R. E. Orton

Chief Engineer
Acme Steel Co., Chicago

Part VIII

Contact Loading on Balls and Rollers

CONTACT loads and the stresses in their vicinity have been frequently discussed in the past. In these references it has been pointed out that a concentrated load is only a theoretical consideration and that in actual fact no load can be concentrated but must be distributed over some area, although this area may be small. At distances remote relative to the area, analysis may be made safely on the basis of a concentrated load; but at points close to the load it is necessary to consider the actual load distribution. Since the stresses in this region may be high, frequently being the determinant of failure, their analysis is of paramount importance.

Original solution of the general problem of contact loading is credited to H. Hertz in 1881. Since then it has been applied to numerous two-dimensional problems. It is the fundamental theory behind the design of rollers and raceways in roller bearings and has been applied successfully to the design of cams, pawls, and other similar parts. It also made possible the first scientific attack on the problem of the wear life of gear teeth, as presented in the works of several authorities on gearing[1,2,3].

The problem of contact loading was solved by Hertz[4] as three-dimensional, with the roller treated as a special case. His theory is presented in some of the books on elastic theory[5,6]. The most complete mathematical derivation that has come to the writer's attention is that of Hoersch[7]. Here again the development is in three dimensions with two dimensions treated as a special case.

Is Two-Dimension Development

Presented here is the development in two dimensions directly—present article covering the equations for the load distribution. The next article will analyze the stresses along the line of roller centers and will follow with curves and data to facilitate the ready and rapid solution of problems. While the intention is to have the text following the derivation stand complete in itself, it is recommended that the mathematics be followed, at least for the purpose of observing the approximations, of which there are a number.

Development of the solution is for two rollers, a plane surface being considered as a roller with an infinite radius, and a circular seat as a roller with a negative radius. Rollers must have sufficient length to develop plane strain. Solutions of previous problems have been the same for plane stress as for plane strain. Here, however, the load distribution is dependent on distortion, which necessitates the limitation. The length, however, need be long relative only to the width of contact, which may be and usually is met by rollers which are short relative to their diameters.

Illustrated in *Fig.* 78 are two unloaded rollers with parallel axes in contact along their length. The radius of the upper roller is r_1 and of the lower

[1] *Spur Gears* by Earle Buckingham, McGraw-Hill, 1928 1st edition, Pages 304 to 309.
[2] "The Design of Helical Gears" by W. P. Schmitter, MACHINE DESIGN, June and July, 1934.
[3] *Recommended Practice of the A.G.M.A. for Metal Spur Gears* adopted May, 1934.
[4] Hertz actually solved only for the loading, and not for the subsurface stress.
[5] *Mathematical Theory of Elasticity* A. E. H. Love, Cambridge Press, 1927.
[6] *Theory of Elasticity* S. Timoshenko, McGraw-Hill, 1934.
[7] *Stresses Due to the Pressure of One Elastic Solid Upon Another* by Howard R. Thomas and Victor A. Hoersch, University of Illinois Engineering Experiment Station *Bulletin* No. 212, 1930.

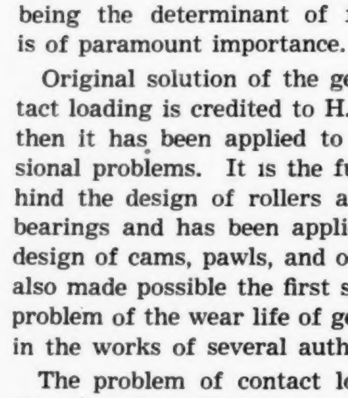
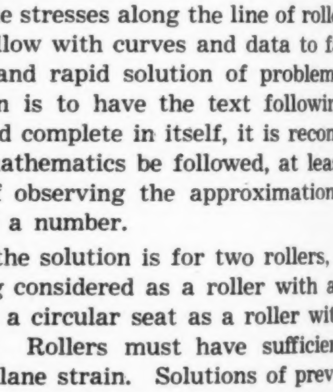


Fig. 78—Two rollers with parallel axes in contact, unloaded

Fig. 79—Two rollers in loaded contact illustrating deformation of surface



actical Design

r_2. Elastic constants are similarly noted by subscript. M_1 is the contact "point" on the upper roller, M_2 on the lower, the two points being at this time common. Different coordinate axes are used for each roller. The Y axis for the upper passes from M_1 through the roller center, and that for the lower from M_2 in the reverse direction. The X axes pass also through M_1 and M_2, respectively, the two axes being at this time coincident.

Application of load to the rollers will widen the line of contact into a narrow rectangle over which the load will be distributed in some fashion. The loading will cause a deformation of the surface of the rollers which will be, in the main, local. This then, will permit the rollers as a whole to approach each other a small amount. The total movement of points within or close to the loaded area will then depend upon the movement of the rollers as a whole, and upon the local strain.

The above is illustrated in *Fig.* 79. To demonstrate better what occurs, the deformation and contact width have been greatly exaggerated. Actually, in the usual case, this deformation will be small. In fact, as will be shown, the equations will not apply to as large a strain as this. The unstrained positions of the contact points, M_1 and M_2, are now separated by an amount e. Their strained positions, M'_1 and M'_2, of course remain in contact, the sum of their strains equalling e. The X axes for the two rollers, since they pass through the unstrained positions, will also be separated by e.

Load Distribution Has Definite Pattern

The distance separating any other pair of opposite points after loading is a function of the position of the points on the rollers before loading, the movement of the rollers as a whole, and the local strain. At points outside the contact surface this distance may theoretically have any value; but at points within the surface it must be zero to maintain continuous contact. The only variable in this distance is the strain, and since the strain will be a function of the load distribution the distribution must follow a definite pattern to fulfill this condition. Therefore, by establishing an expression for the distance originally separating any two points and one for the strain within the contact area, an equation is developed that the load distribution must satisfy.

In *Fig.* 80 is illustrated an undeformed roller. R

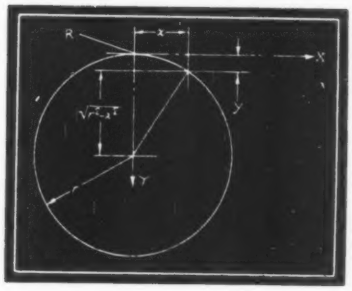


Fig. 80—Location of a point on the surface before deformation

is a point on its surface for which the distance y to the X axis is to be found in terms of the radius r, and x. From the diagram

$$r = y + \sqrt{r^2 - x^2}$$

Clearing of the radical and rearranging

$$y = \frac{x^2 + y^2}{2r}$$

The first approximation appears in the simplifica-

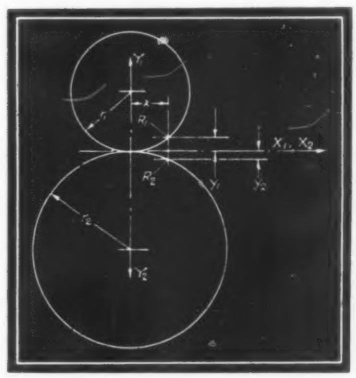


Fig. 81 — Distance between two opposite points before loading

tion of the above. If x is small relative to the radius r, y will be very small relative to x and the quantity y^2 may be ignored. Therefore

$$y = \frac{x^2}{2r} \quad \text{.................................(134)}$$

As an example of the approximation, if r is larger than $10x$, the error in the use of Equation 134 will be less than one quarter per cent.

In *Fig.* 81, R_1 is a point on the surface of the up-

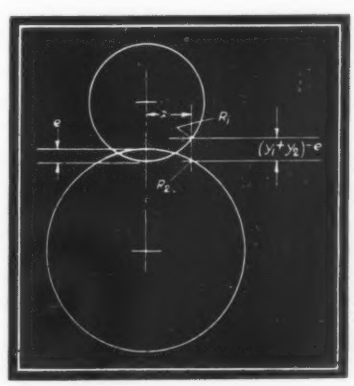


Fig. 82—Hypothetical condition of rollers pressed together without undergoing any strain

per roller and R_2 the corresponding point on the lower roller. The distance between them will be y_1 plus y_2, which, from Equation 134 is

$$y_1+y_2=\frac{x^2}{2r_1}+\frac{x^2}{2r_2}=\frac{1}{2}\left(\frac{1}{r_1}+\frac{1}{r_2}\right)x^2=\beta x^2 \quad (135)$$

where β, known as the "relative curvature," is

$$\beta=\frac{1}{2}\left(\frac{1}{r_1}+\frac{1}{r_2}\right) \quad (136)$$

If the rollers could now move toward each other the amount e, without undergoing any strain, they would occupy the interfering positions shown in *Fig.* 82. R_1 and R_2 would now be apart an amount $(y_1+y_2)-e$. If the value of x is such as to bring these two points within the region of interference this quantity will be negative. The strains, then, must be such as to return this quantity to zero.

Letting u_1 represent the strain of R_1 in the positive direction of the Y_1 axis, and u_2 similarly for R_2, these strains will increase the distance between the two points by an amount equal to their sum. After straining then, as shown by *Fig.* 83, this distance will be $(y_1+y_2)-e+(u_1+u_2)$.

If x is such as to bring R_1 and R_2 within the contact surface, this resultant distance is to equal zero. Therefore, substituting Equation 135 for (y_1+y_2)

$$u_1+u_2=e-\beta x^2 \quad (137)$$

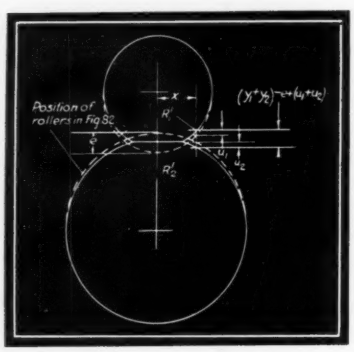


Fig. 83—Condition of rollers pressed together and strained

Now expressions for u_1 and u_2 in terms of the load distribution, substituted in 137, will give the equation that the distribution must satisfy. There is, however, no solution available from which the deformation in a roller may be obtained. It is not feasible to set it up from the solution for concentrated loads on a circular disk because these equations are too involved to lend themselves to the required integration. This leads to the next approximation—the obtaining of the strain from the solution for the semi-infinite plate.

The use of this solution is based on the assumption that the width of contact is so small relative to the radius of curvature that, for the purposes of determining the strain in this region, it may be assumed that the surface of the roller is flat. This approximation might be compared to that of ignoring the effect of the radius of the earth when constructing a map of a small region. Since the width of the loaded area will be as small as two per cent or less of the diameter, this is equivalent to a map of an area 160 miles or less in extent. If conditions should be such as to make this contact quite wide, the solution would not apply.

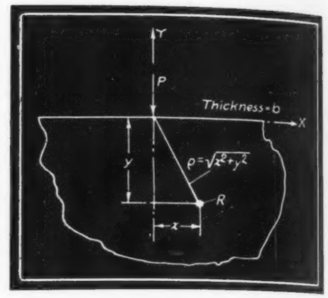


Fig. 84 — Concentrated load on a semi-infinite plate

A concentrated load applied normally to the boundary of a semi-infinite plate is shown in *Fig.* 84. Equations[8] 109, 110 and 111, which are repeated below, give the stress at any point R within the plate.

$$S_x=\frac{2Px^2y}{\pi b\rho^4} \quad (109)$$

$$S_y=\frac{2Py^3}{\pi b\rho^4} \quad (110)$$

$$v_{xy}=\frac{2Pxy^2}{\pi b\rho^4} \quad (111)$$

The plate is here considered to have sufficient "thickness" to develop plane strain. Equation 4[9] gives the relationship between the total strain and the unit strain. Part II of this series gives the relationship between the unit strain and the stresses[10] for plane strain. Substituting in Equation 4 gives

$$\frac{\partial u_x}{\partial x}=\frac{1+\lambda}{E}[S_x-\lambda(S_x+S_y)]$$

$$\frac{\partial u_y}{\partial y}=\frac{1+\lambda}{E}[S_y-\lambda(S_x+S_y)]$$

Substituting the stresses from Equations 109 and 110 gives

$$\frac{\partial u_x}{\partial x}=\frac{2P(1+\lambda)y}{\pi bE}\left(\frac{x^2}{\rho^4}-\frac{\lambda}{\rho^2}\right)$$

$$\frac{\partial u_y}{\partial y}=\frac{2P(1+\lambda)}{\pi bE}\left(\frac{y^3}{\rho^4}-\frac{\lambda y}{\rho^2}\right)$$

Substituting $\rho=(x^2+y^2)^{1/2}$ and integrating

$$u_x=\frac{P(1+\lambda)}{\pi bE}\left[\frac{-xy}{x^2+y^2}+(1-2\lambda)\arctan\frac{x}{y}\right]+f(y) \quad (138)$$

$$u_y=\frac{P(1+\lambda)}{\pi bE}\left[(1-\lambda)\log(x^2+y^2)+\frac{x^2}{x^2+y^2}\right]+g(x) \quad (139)$$

[8] MACHINE DESIGN, July, 1941, Page 62.
[9] MACHINE DESIGN, February, 1941, Page
[10] MACHINE DESIGN, March, 1941, Page 39.

in which $f(y)$ is a function of y only, and the $g(x)$ of x only.

Equation 5 gives the shear strain, γ_{xy}, in terms of the normal strains u_x and u_y. Equation 17 gives the shear strain in terms of the shear stress, v_{xy}. Equating and substituting Equation 111 for the shear stress

$$\frac{\partial u_x}{\partial y} + \frac{\partial u_y}{\partial x} = \frac{4P(1+\lambda)xy^2}{\pi bE\rho^4}$$

Differentiating Equation 138 with respect to y, and 139 with respect to x, and substituting in the above

$$\frac{4P(1+\lambda)xy^2}{\pi bE\rho^4} + \frac{d}{dy}f(y) + \frac{d}{dx}g(x) = \frac{4P(1+\lambda)xy^2}{\pi bE\rho^4}$$

Since $f(y)$ may contain no terms in x, and $g(x)$ none in y, their derivatives must each be equal to zero to satisfy the above. If the derivatives are equal to zero the functions themselves must be constant, therefore

$$f(y) = C_1\ , \quad g(x) = C_2$$

The value of these constants will depend upon the points in the plate that are considered as fixed. Fixing the Y axis, that is, making $u_x = 0$ when $x = 0$, gives $C_1 = 0$. To evaluate C_2 it may be assumed that some point on the finite boundary is fixed in a vertical direction. That is, when $y = 0$ and $x = C$, where C is any constant, $u_y = 0$. Substituting in Equation 139 gives for C_2

$$C_2 = -\frac{P(1+\lambda)}{\pi bE}[(1-\lambda)(\log C^2)+1]$$

Substituting this value of C_2 in Equation 139 and setting $y = 0$, gives for the deformation, in a vertical direction, of the finite boundary

$$u_y = -\frac{P(1-\lambda^2)}{\pi bE}(\log C^2 - \log x^2) \ldots\ldots\ldots\ldots (140)$$

In Fig. 85 is shown the semi-infinite plate under a load of varying intensity q, distributed over the boundary from $-a$ to a. We are interested in the deformation of a point R located on the boundary at a distance x from the origin, and lying within the loaded area, that is $-a \leq x \leq a$. The deformation at R, due to an element of the load at a distance

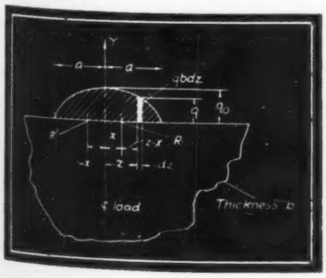


Fig. 85—Distributed load on a semi-infinite plate

z from the origin, is given by Equation 140 if P is replaced by $qbdz$, and x by the distance from R to the load element, that is $(z - x)$, thus

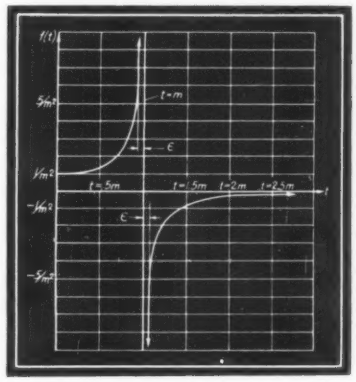
$$\Delta u_y = -\frac{q(1-\lambda^2)dz}{\pi E}[\log C^2 - \log (z-x)^2]$$

The summation of these elements of strain gives

$$u_y = -\frac{1-\lambda^2}{\pi E}\left[\int_{-a}^{a} q \log C^2 dz - \int_{-a}^{a} q \log (z-x)^2 dz\right] \ldots\ldots\ldots\ldots (141)$$

The value q in the above is a function of z only.

Fig. 86—Graph of the function $f(t) = 1/(m^2-t^2)$



In the integration x is constant. C should be measured from the elements of the load and would therefore be a variable. However, if C is made very large compared with a, it may without sensible error be measured from the centerline of the load, which would make it a constant.

Values u_1 and u_2 may now be obtained from Equation 141 and substituted in Equation 137 to give a differential equation that the distribution of the pressure q must satisfy. The following will prove that an elliptical distribution as given by

$$q = \sqrt{a^2 - z^2}\left(\frac{q_0}{a}\right) \ldots\ldots\ldots\ldots (142)$$

(where q_0 is the intensity at the center) will satisfy Equation 137.

Substituting Equation 142 in the first integral of Equation 141

$$\int_{-a}^{a} q \log C^2 dz = \frac{q_0 \log C^2}{a}\int_{-a}^{a}(\sqrt{a^2-z^2})dz$$

the value of which, integrated between the limits, is

$$\int_{-a}^{a} q \log C^2 dz = \frac{1}{2}\pi a q_0 \log C^2 \ldots\ldots\ldots\ldots (143)$$

The second integral in Equation 141 may be
(*Continued on Page 126*)

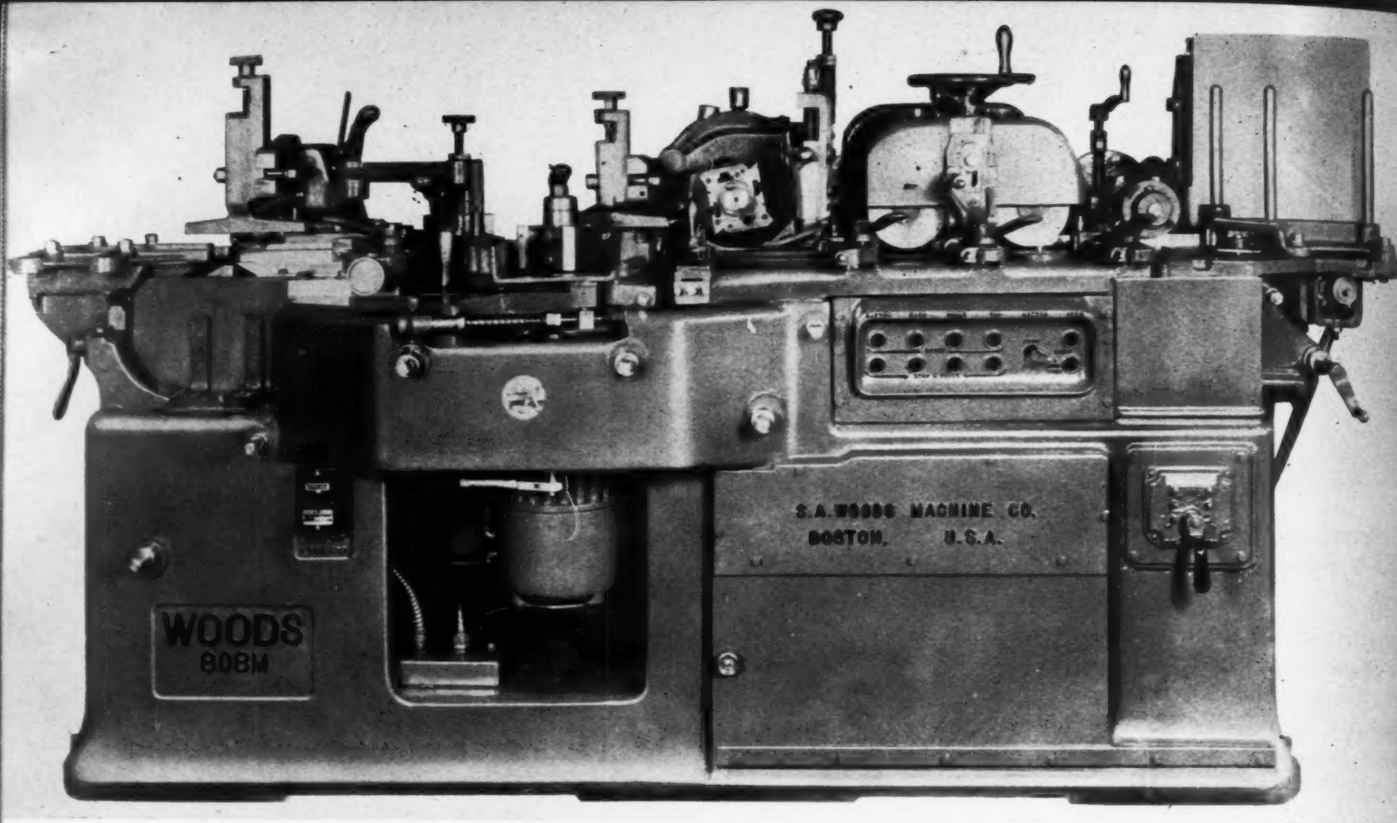


Fig. 1—High speed machine utilizes shell type motors for each of four spindles

Compact Spindle Design Aids High-Speed Operation

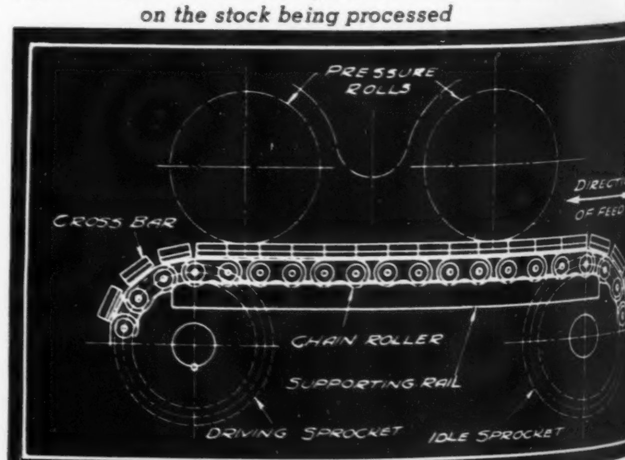
By Lewis B. Carlson
Engineering Department
S. A. Woods Machine Co.

TODAY, as never before, engineers in all fields are being called upon to design machines that will give greater production with as high, or higher, quality of product. To meet this challenge in the woodworking industry a small, high capacity electric molder, *Fig.* 1, capable of planing lumber into miscellaneous shapes was developed. Much thought was given to high-speed spindle mountings, drives, accessibility, machine assembly, and ease of operation.

Lumber stock enters from the right of the machine and is fed through by two top rolls and a bottom lag bed. It can be fed automatically by a hopper feed as shown, or by hand. The hopper feed is used for short lengths of stock.

Individual control of all the motors is furnished by a pushbutton panel which includes a master safety switch. A multispeed feed motor drum is mounted in the frame, together with magnetic starters for the spindle motors. Another pushbutton station for the feed motor is located at the feed-out end of the machine. Starters are solenoid type to give overload and low voltage protection to each motor. All wiring is concealed within the base except for the loops necessary for the movement of the motors. Motors are totally enclosed, fan-cooled, squirrel-cage induction type, specially de-

Fig. 2—Lag bed feed is designed to reduce unit pressure on the stock being processed



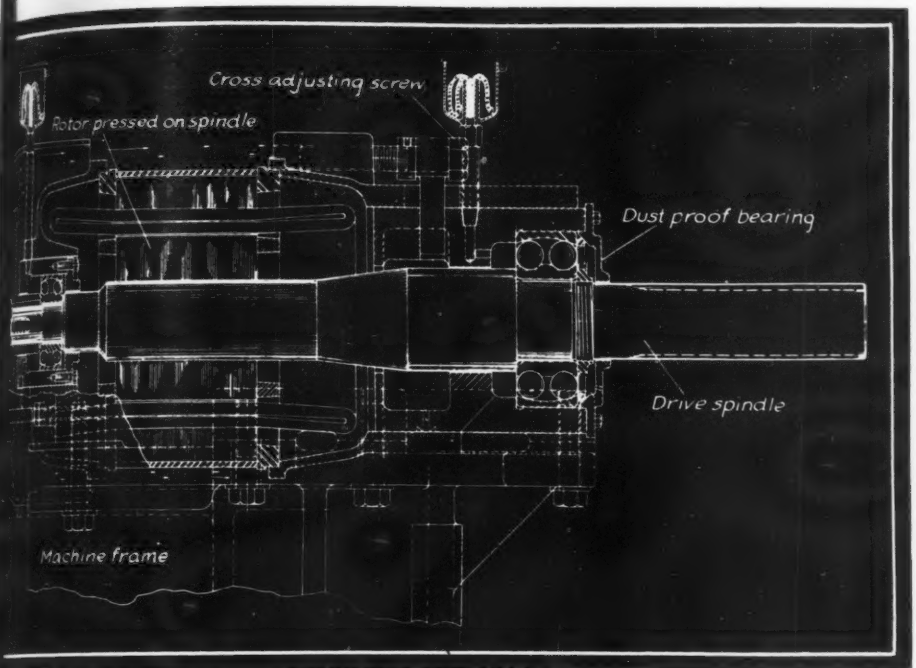


Fig. 3—Top and bottom horizontal spindle assembly. Bearing housing is dustproof

signed for woodworking tools. All cutting spindles are provided with electric brakes controlled by the individual pushbuttons. Top roll shafts are mounted on roller bearings housed in swing yokes. These yokes can be adjusted for varying thicknesses of stock by a handwheel located just above the rolls.

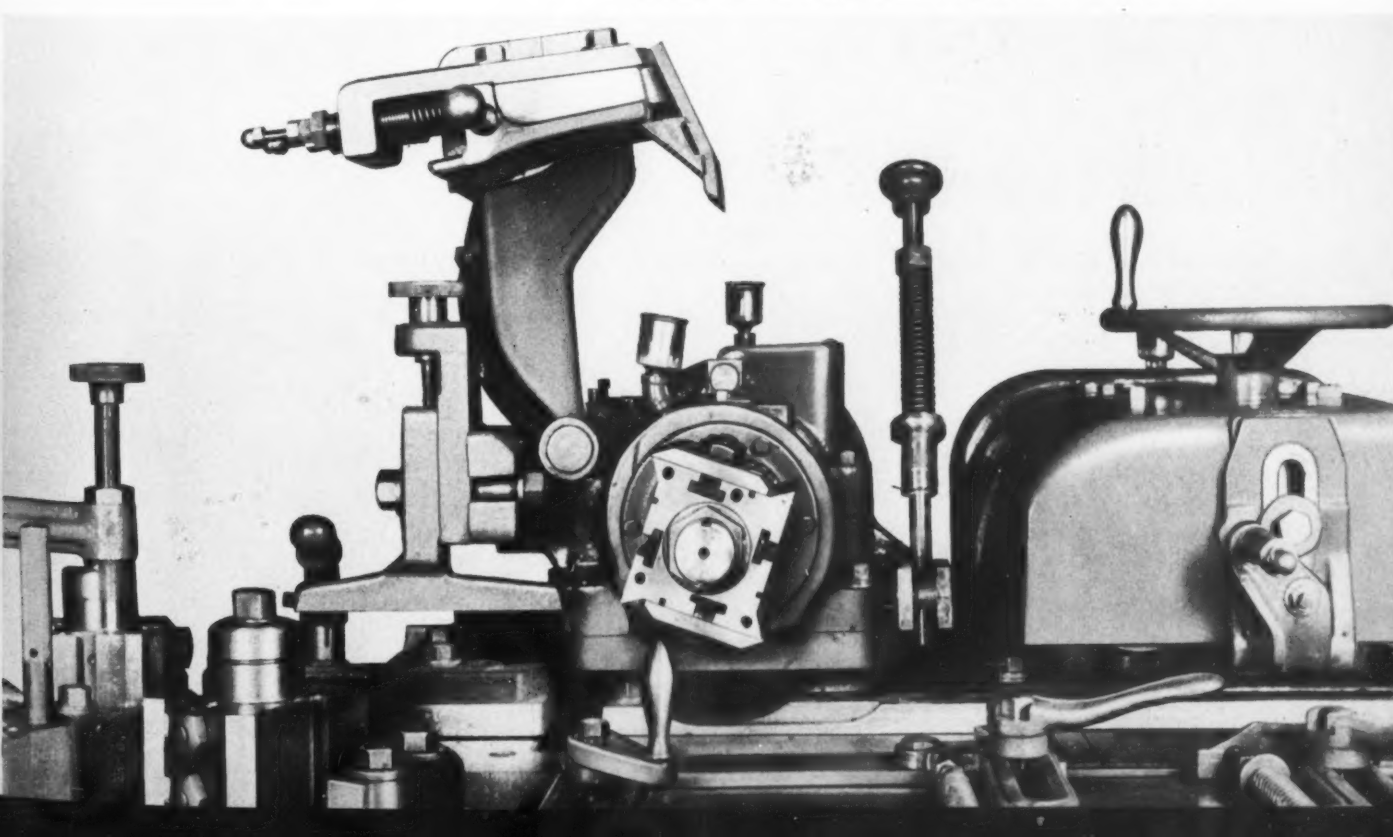
Directly beneath the rolls is a lag bed feed, shown in *Fig.* 2, which is made up of triple roller chain with corrugated cross bars riveted to the links. The chain is supported on two hardened steel rails mounted in a heavy bed. Entire lag bed unit is designed so that it can be removed easily for inspection or for the renewal of parts.

Introduction of the lag bed feed has made possible the feeding of stock that heretofore could not be handled. This is because of the reduced amount of roll pressure needed for feeding, thus eliminating any tendency to crush or break the stock. It has been found in practice that raised grain in the finished surface of lumber is traceable to excessive roll pressure.

Special lubricating system for the chain utilizes a large oil reservoir in the front casting, to which is attached a pipe with outlets directly over each strand of chain. Controlled gravity feed assures positive oiling.

Next to the feeding unit is the top cutterhead spindle and chip-breaker. The spindle is mounted in precision ball bearings and operates at 3600 or 7200 revolutions per minute. *Fig.* 3 shows a sec-

Fig. 4—Spindles are designed for accessibility to aid in setting knives. Chipbreaker swings out of way as shown



tional view of the entire spindle and motor mounting.

A feature in the construction of the spindles is that the rotor is pressed on before finishing. Thus the rotor becomes an integral part of the spindle for its final finishing and balancing operation. This construction, together with the use of preloaded bearings shrunk on the spindle, produces the best running spindle possible. The entire spindle unit, with rotor, bearings and housings in place, can be pulled out on the back side of the machine through the motor. Each bearing is provided with filtered wick lubrication. The two vertical spindles and the bottom spindle (located at the feed-out end) have similar ball bearing mountings. All of the cutterhead spindles are high grade, heat treated alloy steel.

Chipbreaker is made in two sections with direct spring pressure for each shoe. *Fig.* 4 shows a close-up view. The shoes are adjustable to and from the head, depending on the knife projection. Entire unit may be uncoupled and swung up for accessibility to the top head when making set-ups. Shoes are steel, heat treated to give maximum wear.

A persistent problem in planing lumber is the occurrence of clipped ends. This is caused by the lifting of the ends of lumber after it leaves the chipbreaker and passes by the cutterheads. For this reason the chipbreaker, which is used to hold the stock against the machine bed, is equipped with individual yielding shoes extending close to the cutters.

Front and back matcher spindles, vertically mounted, are bolted to heavy matcher plates. Each plate is individually adjustable in machined dovetail ways, and can be clamped positively in position. The front leg is equipped with a swing-out spring tension chipbreaker with sufficient flexibility to take care of varying widths of boards. Entire chipbreaker is removable for knife set-ups.

At the feed-out end of the machine, shown in *Fig.* 5, is the bottom horizontal spindle. This unit is similar to the top, both mounted in sturdy motor yoke castings, and having vertical and cross adjustment.

Because a smooth running spindle depends largely on its mountings and supports, the spindle, with its motor yoke, is mounted on a long, completely guided slide with positive gibbing and clamping, as shown in *Fig.* 6. Each slide is vertically adjustable, sliding up and down in dovetail ways machined in the base.

Initial feed drive from the feed motor which is mounted in the base is also seen in *Fig.* 6. A 3-inch wide belt is used, tightened by a spring tensioned belt binder. Enclosed in an oiltight housing are a pair of helical reduction gears, driving the top roll and lag bed through a roller chain.

Mounting brackets for the chain are provided with adjustment so that the chain can be set with the proper tensioning. To insure long, uninterrupted service, oversize chain has been used in combination with machined steel sprockets.

Bed of the machine is made up of a series of removable plates. Those around the cutting spindles are adjustable too, taking care of varying knife projections. Adjustable hold-down shoes are mounted on heavy bars at the top head, and over the bottom head. Each has screw adjustment and is machined to take a wooden hold-down shoe.

It is only by the use of modern tools, of which this is typical that new methods of manufacture are introduced, thus making possible the greater output so necessary today.

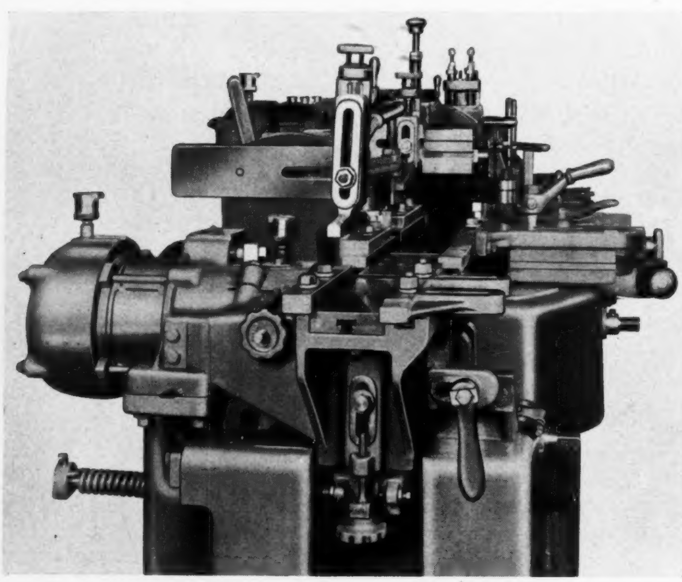


Fig. 5—Above—View from feed-out end shows type of construction and heavy horizontal spindle mounting

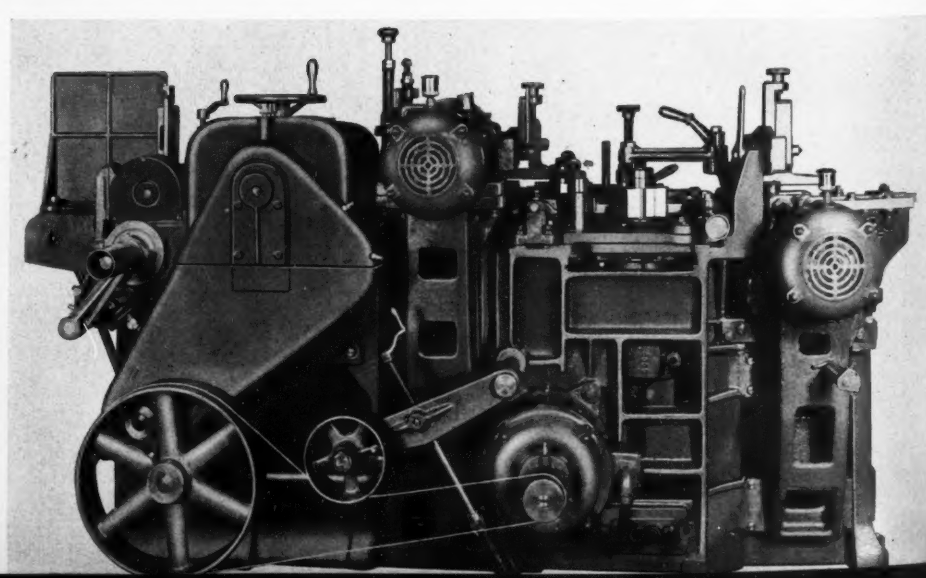


Fig. 6—Right—Back shows one-piece base casting and motor mountings. Webs are on outside to simplify an intricate casting, apperance being of lesser importance here

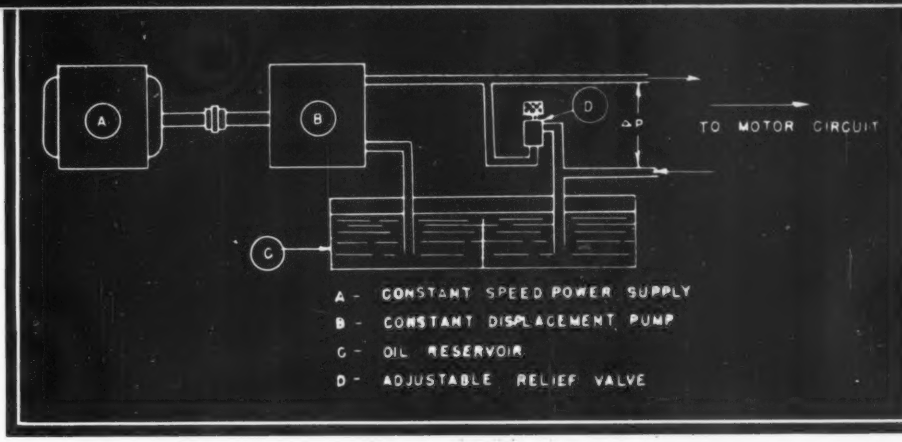


Fig. 1—For low power circuits, relief valve control is satisfactory despite comparative inefficiency

Designing for Hydraulic Torque Control

By
Christian E. Grosser

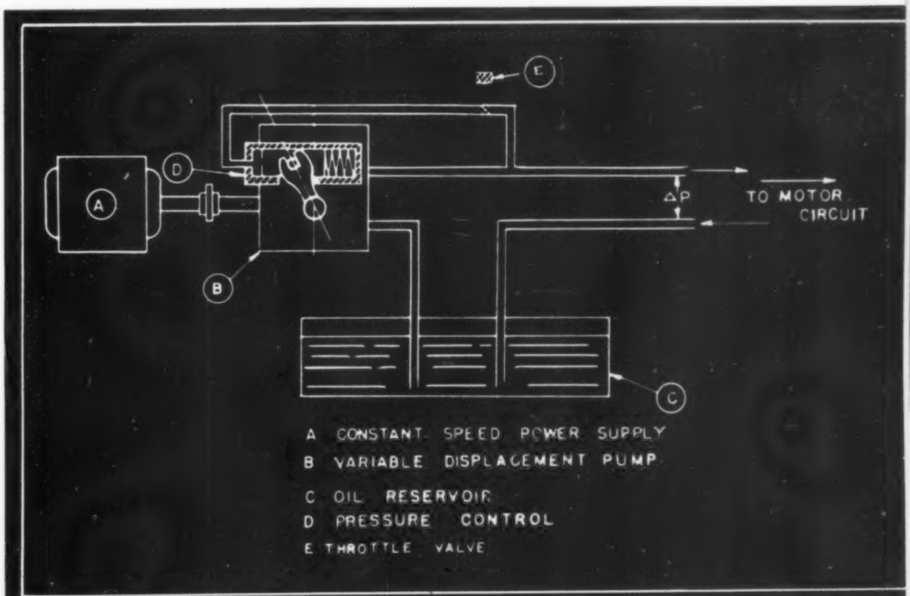


Fig. 2—Approximation of constant pressure control may be obtained by using a control spring with as low a rate as possible

APPLICATION of variable speed to some part of a machine is often desirable in order that it may respond to the motion of some adjoining mechanism and, at the same time, exert a given and constant effort. The effort, a torque or a force, is frequently applied to some material in process, as wire drawn through a die metal strip pulled through reducing rolls, etc. Common examples of such action are recognized in table drives of machine tools, drives for pay-off and take-up reels or capstans, textile warp take-ups, synchronized tandem rolling mill stands, constant acceleration and deceleration mechanisms, paper mill machinery, etc. Customary methods for obtaining such limited torques or forces have been through the use of slipping friction clutches or belts, and electrical torque-type motors.

Friction clutches may introduce problems because of variations in coefficient of friction under changing speed and temperature conditions, difficulties in dissipating the wasted energy of slippage to avoid overheating, wear and replacement.

Friction drives for successive roll stands in a tandem rolling mill are good illustrations of this point. There it is necessary to synchronize a following set of rolls with the preceding set, so that the stock between them will be held at a uniform tension rather than being permitted to sag or be pulled so tightly as to affect the size of rolling, to be stretched or broken. Rolls on succeeding stands are therefore driven through friction clutches at somewhat higher than the stock speed, the stock retarding the rolls to its own speed and causing the friction clutches to slip.

While this method has been acceptable it has never been completely satisfactory because of: The variable torque resistance of the friction clutches which bring irregularities of size into the rolled stock; the necessity of frequent replacement, involving high maintenance costs; and lost machine time on expensive equipment.

Electric torque-type motors function satisfac-

torily in torque applications but have the disadvantage of possessing relatively low torque capacity and often requiring high ratio reduction gears to multiply rotor torque, especially where drive movements are slow. In instances where the torque drive is required to accommodate itself quickly to changes in speed, electric motor rotor inertia is troublesome, resulting in a lag of response accompanied by appreciable changes in the torque or force exerted at the output shaft. While it is possible by elaborate motor controls with "derivative" correction to minimize the effects of rotor inertia, in many applications involving rapid drive response to sudden changes in load position or speed, the drive becomes highly special and consequently expensive.

Hydraulic torque drives have been found to perform satisfactorily in applications such as those listed above, with a minimum of equipment and maintenance.

Hydraulically controlled torques or forces are most conveniently obtained by the use of controlled-pressure circuits. This is obvious in the case where the hydraulic cylinder is used to apply a force, since the force applied by the hydraulic piston is proportional to the liquid pressure in the cylinder. In the case of the constant displacement fluid motor, the output torque is also proportional to the applied liquid pressure by the relationship derived in the July, 1941, article of this series:

$$T = \frac{\Delta p D}{2\pi} \eta_{TM} \qquad (1)$$

where T = shaft torque in inch pounds, p = pressure difference between inlet and outlet in pounds per square inch, D = motor displacement in cubic inches per revolution, and η_{TM} = motor torque efficiency—generally in the range 80 to 95 per cent.

It is convenient to consider controlled-pressure systems as separated into two divisions: (a) the pressure generating apparatus with its controls, energy storage equipment and auxiliaries; (b) the force or torque applying element or elements with their controls and auxiliaries.

Attention is again called to the strong resemblance to the hydraulic that direct current electric circuits possess. This resemblance is particularly close between the conventional constant voltage direct current systems and the constant-pressure liquid circuits.

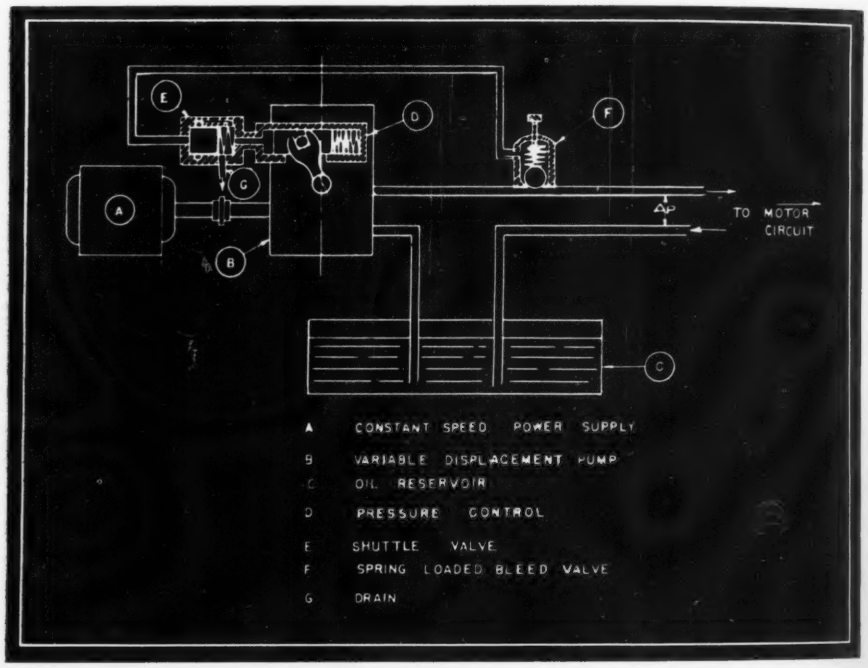


Fig. 3—Shuttle valve permits only slight pressure fluctuations even in the event of sudden changes in flow

An analysis will be made of a variety of constant-pressure generating systems. Following this will be an examination of various motor elements which may be connected to the constant-pressure sources to give the kinds and degrees of torque and force control required for particular machine functions.

Fig. 1 shows the simplest and most widely used arrangement for power drives below about 10 horsepower, where the amount of energy lost by dissipation through the relief valve is not serious due to the adequacy of simple cooling means to dispose of the developed heat, and because operating costs are low.

Probably the most important item is the relief valve which must absorb all the delivery from the pump in the event that flow to the motor end of the circuit is stopped, either by standstill of the motor element or from accidental blocking of the flow. Delivery through the relief valve throughout the complete range from zero to the maximum of the pump should take place with no greater variation

in pressure than the variation tolerated in the force or torque desired at the motor output. The valve should respond quickly to changes in the amount of by-pass, with a minimum of pressure variation during the change in flow.

Ordinary spring-loaded relief valves used for overload protection are seldom suitable for constant-pressure service. On account of internal pressure changes due to the dynamics of the oil flow through the simple ball or piston types, they invariably have the characteristic of a rise in pressure as the flow through them increases. Furthermore, pressure at closing will be found to be different from that at opening. The larger the valve the less severe are such shortcomings, but it is necessary to go to impractical proportions to limit pressure variation to even 25 per cent of the opening pressure in these simple safety relief valves.

Recent developments have produced special valves which in small units may be depended upon to maintain uniform pressure within a few per cent over a wide range of by-pass flow. They may also be readily adjusted to hold various degrees of pressure within widely separated limits. After the permissible pressure variations in the circuit have been established, the designer may determine a suitable valve from a study of the characteristic curves of those commercially available. For extremely exact pressure control it is possible to devise special valves with elaborate pressure-responsive control mechanisms to achieve almost any degree of precision.

It is obvious that a simple orifice, such as a needle valve, can be used as a pressure-controlling means only if the flow in the circuit is uniform.

Oil temperature changes will affect by-pass devices through variation in oil viscosity; in the case of the needle valve, this effect is extreme. The effect on pressure-responsive relief valves will be found to be negligible compared with the effect on the pressure, of the flow in the pipes connecting the motor devices and internal passages in the motors themselves.

Whenever relief valves are used for "dumping" purposes as in the by-pass circuit it is important to discharge the hot oil into a large enough reservoir. This is for the two reasons that sufficient tank area must be exposed to remove the heat under natural cooling, and the oil must rest there long enough so that foam will collapse and not be drawn as entrained air into the suction line of the pump. Foam is created by the release of dissolved air during the sudden transition from high to low pressure in the relief valve, accompanied by increase in temperature. Both influences tend to render the air less soluble in the oil and cause it to appear as bubbles in the discharged oil stream. A baffle between the inlet and outlet pipes in the tank assists in dispersing the foam and has the secondary advantage of permitting dirt particles to settle in the tank and of retarding their repeated circuit through the system.

In all circuits where the major part of the pump delivery may have to be dissipated in heat for extended time intervals, an empirical rule for reservoir capacity is to provide oil storage at least equal to twice the delivery of the pump per minute. This holds for operating pressures below 1000 pounds per square inch. For higher operating pressures the capacity should be proportionately increased.

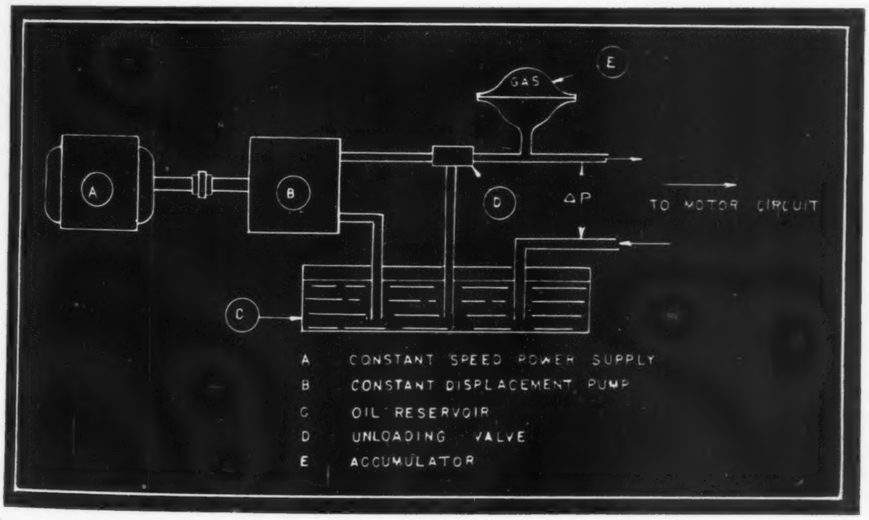


Fig. 4—Either gas or spring type accumulators reduce peak-load demands on drive motor in circuits where load is intermittent

When more than about 20 horsepower are being delivered to a constant-pressure circuit it is usually not economical to use the by-pass method of pressure regulation, nor is it easy to remove the generated heat by natural cooling on account of the large tank necessary.

The better method is to use a variable displacement pump with a pressure-responsive displacement control which will supply oil to the circuit only as it is required to maintain the desired pressure. Consequently no energy is lost except

that due to the operating inefficiency of the circuit itself. Various methods of displacement control may be used depending upon the accuracy of pressure regulation desired. *Fig.* 2 shows a commonly used arrangement for a limited range of pressure settings and where the variation in flow required by the fluid motor is small. Strictly speaking the control shown will theoretically not produce a constant pressure—it is better adapted for power control because, as the displacement of the pump increases it permits a reduction of pressure, since the control piston allows its spring to elongate and reduce the resisting force. However, the control spring may be designed to have a low rate in pounds per inch so that pressure variation may be reduced to perhaps 10 per cent.

One disadvantage of the control is its inherent instability. Unless there is considerable friction introduced into its working parts, or a throttling device such as a small orifice or needle valve placed in the bleed line to the control, the system will hunt, that is, it will permit the system pressure to oscillate above and below the pressure setting.

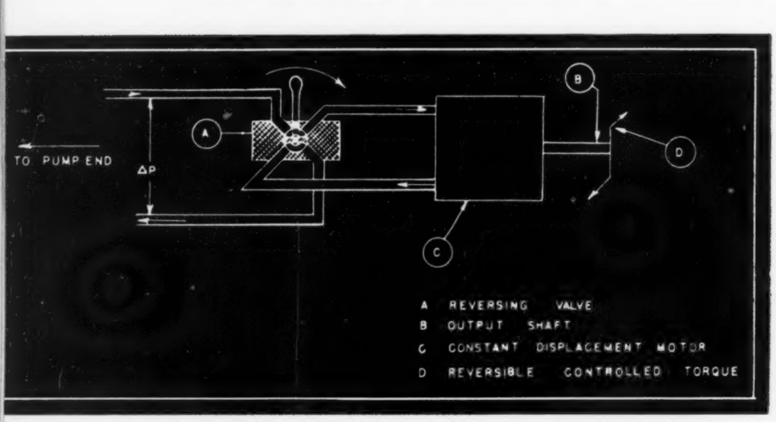


Fig. 5—Reversing valve to change direction of flow may be incorporated in previously illustrated circuits

Fig. 3 shows a control which theoretically permits no variation in pressure with changes in pump displacement. It is very stable in operation and allows no hunting because of the throttling action of the bleed valve. Its response to sudden flow demand changes is better than the circuit in *Fig.* 2, permitting only a slight momentary fluctuation in pressure. Under steady, or slowly varying, circuit flow conditions it may hold pressure to within a few per cent change. The shuttle valve in the bleed line to the control permits release of trapped oil in the control cylinder if the line pressure drops suddenly, requiring instantaneous movement of the displacement control for a larger pump delivery. Under slowly changing flow conditions this shuttle valve is not necessary, since leakage past the control piston permits the trapped oil in the control cylinder to get out. This leakage may be deliberately increased by a small leakage hole through the piston if a higher rate of control response is desired, but the shuttle valve is preferable to this method. The bleed valve is an ordinary spring-loaded relief valve requiring no special construction, for it is required to pass only small amounts of liquid (the volume of the control cylinder) and for these low rates of by-pass its regulation has little chance to vary.

All the above systems of pressure generation may be elaborated by the addition of an accumulator or energy storage reservoir, connected into the high pressure side of the line. This is desirable when the output of the circuit is subject to large variations in speed, as for example with a fluid motor in the form of a hydraulic piston and cylinder used as a pusher or rammer.

Operation of this kind is characterized by large demands of energy over short intervals of time, but with an average demand far below the peaks. The accumulator may be proportioned to have energy enough in reserve to serve the peak demands, and then be recharged by the pump during the intervals when the fluid motor demand is below average. By these means a much smaller pump may be used for pressure generation, with the obvious advantages of lower cost and weight, higher efficiency, and lower maximum input power requirements. Accumulators are of two kinds, spring loaded or gas filled. In the spring-loaded type a heavy cylinder contains a free piston, bearing against a heavy coil spring. At the opposite end of the cylinder high pressure oil is introduced, and as it accumulates in the cylinder the spring is compressed and thus stores energy.

In the gas type the spring is replaced by a space filled with compressed air or an inert gas under pressure equal to the minimum under which the system is to operate. When oil is introduced against the piston, it compresses the gas and thereby stores energy equal roughly to the pressure multiplied by the displaced volume.

Diaphragm Used in Accumulators

The design trend in gas accumulators is toward a spherical construction, with a flexible diaphragm to act as the separating medium between the gas and the oil. The sphere is split on a diameter and the two flanged halves bolted together, with the diaphragm clamped between the flanges.

Unfortunately the accumulator circuit does not lend itself well to applications where the fluid motor output torque or force must be held closely to a given value. By the very nature of the accumulator function to store energy, a considerable variation in pressure is inevitable since the storage of energy is accompanied by a rise of accumulator pressure; discharge of energy conversely brings about a fall of pressure. The magnitude of this rise and fall is somewhat larger in the spring-loading type of accumulator unless a very long spring is used. In the gas type, the

variation is of the order of 20 to 25 per cent. A further disadvantage of the accumulator circuit is that a change in operating pressure cannot readily be made because it involves recharging gas to the new pressure and resetting the unloading valve for a different operating range.

Presence of an accumulator in the circuit makes possible still another kind of pressure-generating arrangement possessing the advantages of the low cost and simplicity of the constant displacement pump, but deriving the benefits of the variable displacement pump through cessation of pressure generation when the circuit has been brought up to pressure. This system is shown in *Fig.* 4.

Pump Starved by Valve

It will be noted in *Fig.* 4 that the unloading valve is connected in parallel with the constant displacement pump. Its internal construction is such as to short-circuit the pump when the accumulator has been charged to a predetermined pressure. The pump delivery is then carried back to the pump suction or to the sump tank, thereby unloading the pump so that its work is simply to circulate oil idly in the by-pass pipes until demands on the system require further pressure generation. When the accumulator pressure falls to a minimum setting, the unloading valve cuts out and the pump resumes duty.

Another type of unloading valve simply cuts off the suction of the pump when it is off duty—"starving" the pump. This latter method, however, has been adapted only to a certain kind of pump which is capable of maintaining pressure with no delivery of oil.

Accumulator circuits are currently operating at pressures near 3000 pounds per square inch, which is near the limit of power hydraulic practice. Appreciable savings in size and weight of equipment are thereby effected.

Turning now to the fluid motor end of torque and force control circuits, there are some broad considerations concerning the choice of the type of fluid motor best adapted to a particular function. If the motion desired at the output is to be rotary and to progress in one direction continuously, as in a reel or mill drive, there is only the obvious choice of the rotary fluid motor. It must be emphasized that by the supply of constant pressure to such a motor, the control of shaft torque under variable speed conditions will be appreciably reduced in effectiveness if the inertia of the motor revolving parts is large. Therefore, it is of primary importance to use fluid motors having a compact construction in applications requiring sensitive tension maintenance. Gear type fluid motors, the axial piston types, and certain special vane motors have low inertia of revolving parts. In some cases of intermittent, reversible, long stroke, translatory motions under a constant force, there is advantage in using a rotary fluid motor to save the space of a long cylinder ram. Examples are in the pulling of long broaches and in pushing of rammers where chains, engaging sprockets driven by the motor shaft, have served admirably; also racks and pinions are used on long planer table drives.

Fig. 5 shows a rotary fluid motor for torque drive with a reverse valve to change direction. This may be attached to any of the pressure supply circuits, *Figs.* 1 to 4. Multiple motors may be supplied by one pressure generator, but a word of caution for this is that if there is any possibility of one fluid motor being unloaded when the other motors are at a standstill, the idle motor may overspeed through having the whole pump delivery applied to it. Overspeeding may be avoided by a hydraulic governor in one of the fluid motor supply lines in the form of a metering valve.

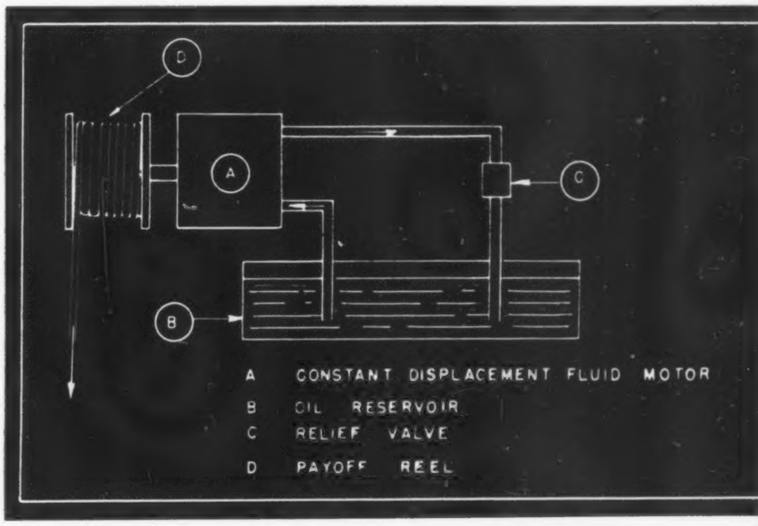


Fig. 6—Torque drag control maintains constant tension in materials between rolls or reels as in textile or rolling mill drives, wire coilers, etc.

It is to be noted that the use of fluid motors with the "open" pressure-generating circuits, as shown in *Figs.* 1 to 4, is restricted only to torque application in the direction of the motion. If "overhauling" loads are likely to be encountered in which the torque may act opposite to the motion, as for example in decelerating a large mass, either a "closed" circuit must be devised with pressure control on both sides of the circuit, or appropriate switching or shuttle valves provided.

Fig. 6 shows a torque drag control such as may be used in a pay-off reel to maintain stock tension on textiles, wire and rope stranding. Where short intermittent translatory motion is required, the hydraulic cylinder is the most efficient fluid motor.

CORRECTION—In the article "Designing for Hydraulic Variable Speed," beginning on page 41 of the July issue, *Fig.* 1 should be *Fig.* 2 and vice versa in order to agree with the manuscript.

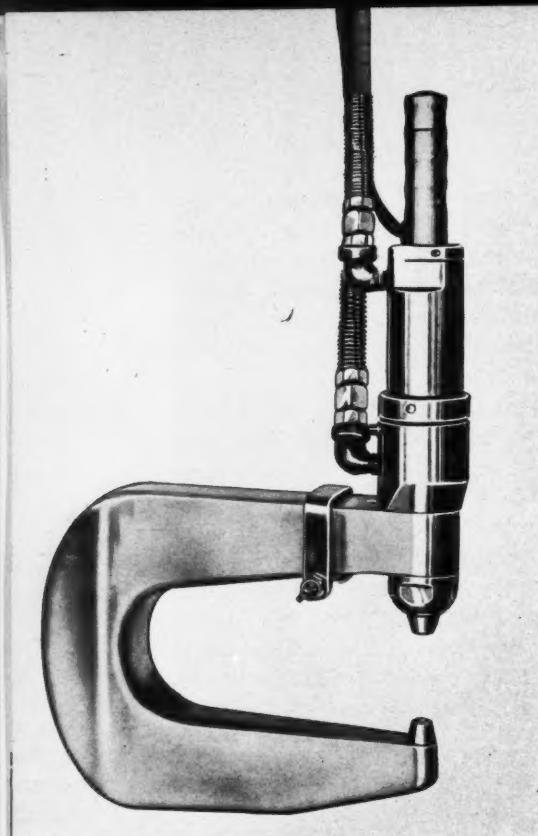


Fig. 1—Riveter typifies the use in machines of C-yokes which require careful calculation of deflection

Avoiding Excessive D

By George S. Hoell
Edw. G. Budd Mfg. Co.

SEPARATION under load of the two free ends of the cantilever arms of C-yokes and the angular displacement between them are of extreme importance in designing yokes such as those shown in *Figs.* 1 and 5. The following is intended as an aid in predetermining both quantities with sufficient accuracy for practical use.

An exaggerated set of lines of elastic deformation is shown in *Fig.* 3. The lines represent approximately the neutral axes of the diagrammatic yoke illustrated in *Fig.* 2. Resultant total deflection at the tip of the cantilevers is made up of several component parts. First, the cantilever itself bends. If it is made with a uniform section and straight sides, the deflection formula for this part alone is readily available. However, the result so obtained should not be taken as the sole factor determining deflection. The difference between the total combined deflection and the deflection of the cantilever alone, can be visualized by comparing the lower and the upper half of *Fig.* 3.

Secondly, the straight back of the yoke marked A will bend backwards and spread the tips considerably, especially if A is great. Lastly, the quarter bend acts in a two-fold way to spread the tips further. In the top half of *Fig.* 3 it is seen that the top of the bend has been lifted up, and the tip of the cantilever will of course be raised the same amount. In addition, the top of the bend has also rotated through an angle θ_3, and the tip of the cantilever will be swung up an additional amount depending upon this angle and the length of the cantilever.

The influence on the deflection of shear forces and direct tension is negligible, and is therefore not considered here. Deflection formulas of (1) the cantilever, (2) the straight back A, and (3) the quarter-bend, follow:

CANTILEVER DEFLECTION: The deflection δ_1 at the free end of a cantilever beam is

$$\delta_1 = \frac{kPL^3}{EI} \quad \text{.......... (1)}$$

and the slope

$$\theta_1 = \frac{k_0 PL^2}{EI} \quad \text{.......... (2)}$$

In these formulas, k and k_0 are constants depending on the character of the beam. *Figs.* 4 and 6 show several types of cantilever

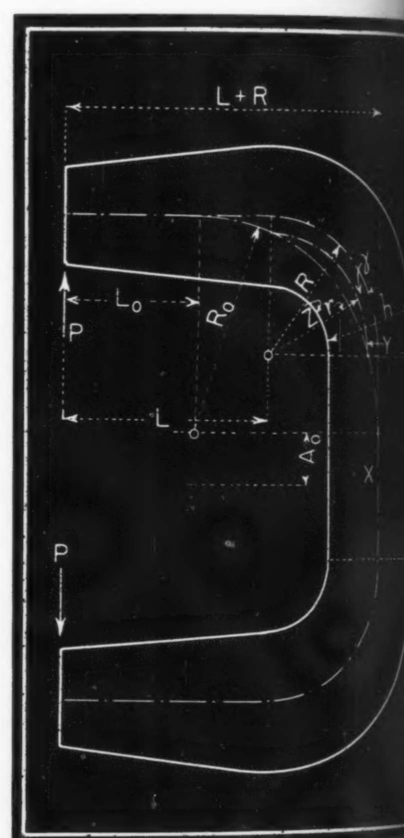


Fig. 2—Dimensions and sections of a typical yoke are used in determining deflections under load. Deflections of neutral axes due to straight section, quarter bends and cantilevers are shown in Fig. 3

Fig. 5—Improper design may result in yoke distortions in machines of this type resembling that shown in Fig. 8

ction in Yoke Design

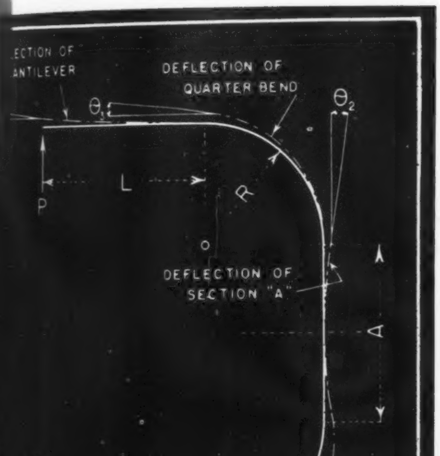


Fig. 3—Dotted lines indicate approximate displacement of the neutral axis of Fig. 2 under load

Fig. 4—Variations of cantilever section of yokes require proper selection of constants from Table I

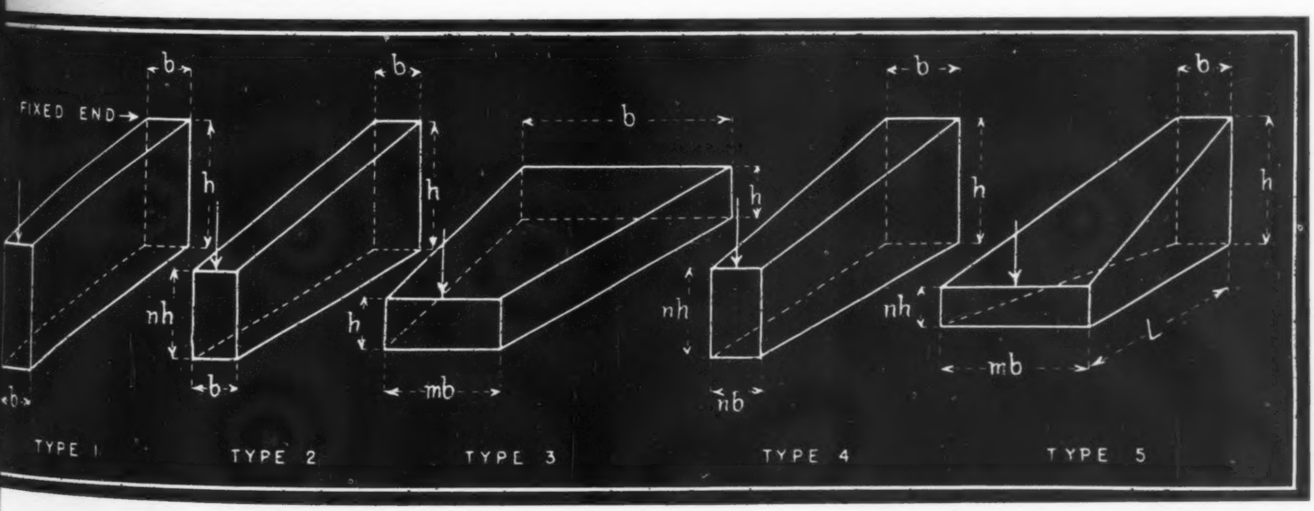
beams. While all possible types cannot be anticipated, those described should be sufficient to cover all usual cases.

TABLE I, giving values of the constants k and k_o, refers generally to rectangular sections where m is the ratio of the width at the free end to the width at the fixed end (which in this case is the end where the beam joins the bend), while n is the ratio of the depth at the free end to the depth at the fixed end. However, in the case of type 4, *Fig.* 4, where the ratios m and n are equal, the tabulated constants k and k_o hold also for any cross section whatever, as long as all dimensions taper at the same rate.

The tables have been computed for a few ratios of m and n. For other ratios sufficient accuracy will be obtained by interpolation. The types shown in *Fig.* 4 exemplify the physical proportions corresponding to the various ratios of m and n.

These types all have straight edges, and comparatively simple formulas can be given for types 2, 3 and 4. The method of finding the deflection for type 5 *Fig.* 4, also applies to the irregularly changing beam shown in *Fig.* 6. The method is an approximation, but when applied to cases for which the accurate formulas are given, and within the scope of the dimensional proportions here dealt with, the results check within .001.

Consider the beam divided in a number of parts of equal length. Compute the moment of inertia at the midpoint of each part with respect to the neutral axis of the cross section, and call these $I_1, I_2, I_3, \ldots I_p$, p being the number of parts in which the beam is considered di-



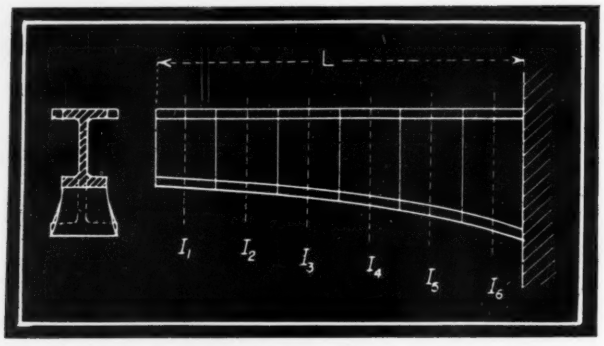


Fig. 6—Cantilevers with other than straight edges can be calculated by the method of summations

vided and I_p the moment of inertia of the midsection of the part nearest the fixed end.

Deflection at the free end of such a beam will then be

$$\delta_1 = \frac{PL^3}{4EP^3}\left[\frac{1}{I_1} + \frac{3^2}{I_2} + \frac{5^2}{I_3} + \ldots\ldots \frac{(2p-1)^2}{I_p}\right] \ldots\ldots(3)$$

The slope at the free end of the beam will be

$$\theta_1 = \frac{PL^2}{2Ep^2}\left[\frac{1}{I_1} + \frac{3}{I_2} + \frac{5}{I_3} + \ldots\ldots \frac{(2p-1)}{I_p}\right] \ldots\ldots(4)$$

Deflection Due to Straight Section: This section, *A*, will bend outwardly under a spreading load at the tips. The angle through which each end bends is

$$\theta_2 = \frac{PA(L+R)}{2EI} \ldots\ldots(5)$$

Corresponding displacement of each tip of the cantilever will be

$$\delta_2 = \frac{PA(L+R)^2}{2EI} \ldots\ldots(6)$$

Deflection Due to Quarter Bends: The quarter-bends are subject to the loading condition shown in *Fig.* 7. On the assumption that within the small deflections encountered in practice the deflection under the load *P* is proportional to the load, the deflection will be, for each quarter bend

$$\delta_3 = \frac{PR}{2EI}\left(\pi L^2 + 4LR + \frac{\pi R^2}{2}\right) \ldots\ldots(7)$$

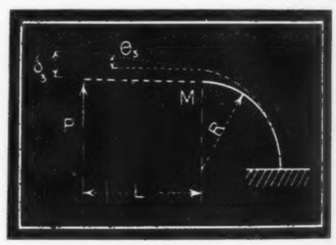
and the angular displacement at the tip of the cantilever due to flexure of the quarter bend

$$\theta_3 = \frac{PR}{2EI}(\pi L + 2R) \ldots\ldots(8)$$

Use of the radius *R* in these formulas is safe for practical use because the result will be somewhat greater than actual. According to the Curved Beam theory the radius of the neutral axis lies on a circle somewhat inside the axis through the center of gravity of the section. This, however, applies only to a bend of constant curvature with no straight ends attached. The smaller radius mentioned is indicated by *r* in *Fig.* 2, and *Fig.* 7. Actual neutral axis is modified under the influence of the straight ends to a curve indicated in *Fig.* 2 by the radius R_0. This condition is shown more vividly in *Fig.* 9 which represents a celluloid specimen under plane polarized light. This model was subjected to a constant bending moment to avoid the influence of shear stresses. The heavy central line marked with a radius R_0 is the actual neutral axis, and an accurate calculation of deflection would have to take this condition into account. This is not necessary, however, for the degree of accuracy which a designer would require in a practical case. While the condition has comparatively little influence on the total deflection, it has a decided influence on the stresses at the inside of the bend, marked *Z* in *Fig.* 2. While this article is primarily concerned with deflections, a discussion of the stress condition at this point is appended because of its importance in designing structures of this class.

For the purpose of showing the manner in which

Fig. 7 — Assuming small deflections, displacement of the quarter bends may be considered proportional to load



the foregoing equations may be used, the following example is cited.

Let the yoke shown in *Fig.* 10*b* have a uniform cross section as shown in *Fig.* 10*a*, except that the lower cantilever is tapered so that all dimensions at the tip are two-thirds of the dimensions of the section shown.

Deflection constants *k* and k_o are found from Table I to be:

For the upper arm $k = .333$, $k_o = .500$

For the lower arm $k = .500$, $k_o = .875$

Moment of inertia of the section *Fig.* 10*a* is computed to be 22.08, the distance from the inner edge to the center of gravity is 2.071 inches, making $R = 8.071$. Using this value of *R* in the formulas, and assuming a load of 5000 pounds, we have for steel ($E = 29{,}000{,}000$)

$$\frac{P}{EI} = \frac{5000}{29\times10^6\times22.08} = .0000078$$

Deflection of upper arm

$$\delta_1 = kPL^3/EI = .0000078 \times .333 \times 20^3 = .0207$$

Deflection of lower arm

$$\delta_1 = kPL^3/EI = .0000078 \times .500 \times 20^3 = .0312 =$$

Spreading of both tips due to bending of section A (see *Fig.* 3)

$$2\delta_2 = 2PA(L + R)^2/2EI = .0000078 \times 4 \times (20 + 8.071)^2 = .0246$$

Spreading of both tips due to deflection of both quarter bends

$$2\delta_3 = 2PR[\pi L^2 + 4LR + \pi R^2/2]/2EI = .0000078 \times 8.071 \times (3.14 \times 20^2 + 4 \times 20 \times 8.071 + 1.57 \times 8.071^2) = .1260$$

Total deflection .... .2025

It is seen that in this case the flexure of the quarter bends accounts for the greater part of the deflection.

The corresponding angular displacements of the tips will be

Rotation of upper arm at tip, in radians

$$\theta_1 = k_0PL^2/EI = .0000078 \times .500 \times 20^2 = .00156$$

Rotation of lower arm, at tip

$$\theta_1 = k_0PL^2/EI = .0000078 \times .875 \times 20^2 = .00273$$

Due to deflection of both ends of section A

$$2\theta_2 = 2PA(L + R)/2EI = .0000078 \times 4 \times (20 + 8.071) = .00088$$

Due to deflection of two quarter bends

$$2\theta_3 = 2PR(\pi L + 2R)/2EI = .0000078 \times 8.071 \times (3.14 \times 20 + 2 \times 8.071) = .00498$$

Total angle between axes of tips (radians). .01015

TABLE I

Table of Deflection Constants k and k_o

m (ratio of width of free end to fixed end)	n (ratio of depth of free end to fixed end) 1		3/4		2/3		1/2		1/3	
	k	k_0	k	k	k	k_0	k	k_0	k	k_0
1½	.295	.431	.367	.566	.390	.634	.479	.829	.600	1.218
1	.333	.500	.411	.667	.447	.750	.545	1.000	.708	1.500
3/4	.356	.548	.444	.741	.485	.845	.597	1.149	.781	1.777
2/3	.366	.567	.457	.776	.500	.875	.619	1.183	.821	1.888
1/2	.386	.614	.493	.860	.535	.975	.667	1.333	.899	2.162
1/3	.409	.676	.525	.966	.579	1.122	.732	1.565	1.000	2.500
0	.500	1.000	.669	1.563	.784	1.872	1.020	2.986	1.664	5.824

This angle, in degrees, is 57.3 × .01015 = .582 degrees. The angle is not great, but if the yoke were used as shown in the welding gun, *Fig.* 8, the contact points of the welding tips would be offset about 7/32-inch when the pressure is applied.

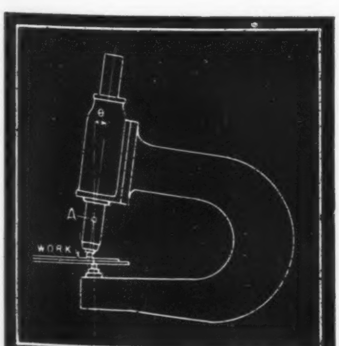


Fig. 8—Cumulative deflections of straight sections, quarter bends and cantilevers may result in faulty operation of a machine

This displacement as well as the deflection of more than 3/16-inch would indicate a yoke too weak for the load specified, even though permissible stresses may not be exceeded.

In actual practice a designer would probably shape the back of the yoke on a line G — H, *Fig.* 10*b*. In many such cases the solution of the deflection problem is too involved for practical use and the designer must often be satisfied with the knowledge that, compared with a simplified case, the deflection as well as stresses of the structure under consideration will be less than the permissible limits. In other words, judgment must be used in simplifying each case, so that the actual structure will be stronger and stiffer than calculations for the simplified case would indicate.

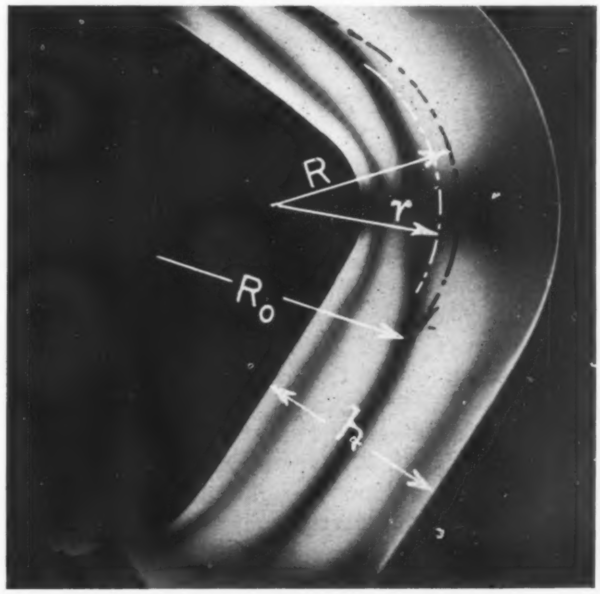


Fig. 9—Photograph of photoelastic model indicates displacement of the neutral axis under load

Cantilevers Affect Shift of Axis

STRESS CONCENTRATIONS AT BENDS: In *Fig.* 9, was shown how the neutral axis changed its course due to the combination of a bend with two straight ends. The cross section at point Z, *Fig.* 2, must resist the applied bending moment by internal stresses in the small space between the actual neutral axis and point Z on one side, and distributed more freely on the other side of the neutral axis out to the outer edge of the bend. Naturally the stresses at point Z must be greatly increased beyond those of a straight section having its neutral axis through the center of gravity.

From the elementary Curved Beam theory the stress at point Z may be determined from the equation

$$s_1 = \frac{M\left(\frac{h}{2} - \gamma\right)}{F\gamma R_i} \qquad (9)$$

in which M is the bending moment, F the area of the section, and γ the displacement of the neutral axis, the other symbols being those indicated in *Fig.* 2. Since γ is dependent upon the curvature of the outside and inside edges of a rectangular section, the new increased γ found from the photoelastic specimen cannot be substituted in the formula for s_1. The modified radii of curvature which would produce a displacement equal to that measured must first be calculated. On the basis therefore of a measured γ, the ratio α between the outside and inside modified radii can be found from the expression

$$\frac{\gamma}{h} = \frac{\alpha+1}{2(\alpha-1)} - \frac{1}{\log_e \alpha} \qquad (10)$$

This equation is not explicit with respect to α but after using a few trial values the real value can soon be determined graphically. Having the ratio α between the outside and inside modified radii, the inside modified radius can be determined, since the depth h of the section is known. With these new values of γ and R_i, the stress concentrated at Z can be determined from Equation 9 given on the previous page.

Fig. 11 is a diagram indicating the stresses over a section of depth h. Stresses s, s_1 and s_0 are tension stresses, those on the right side of the diagram being compression stresses. The stress s indicates

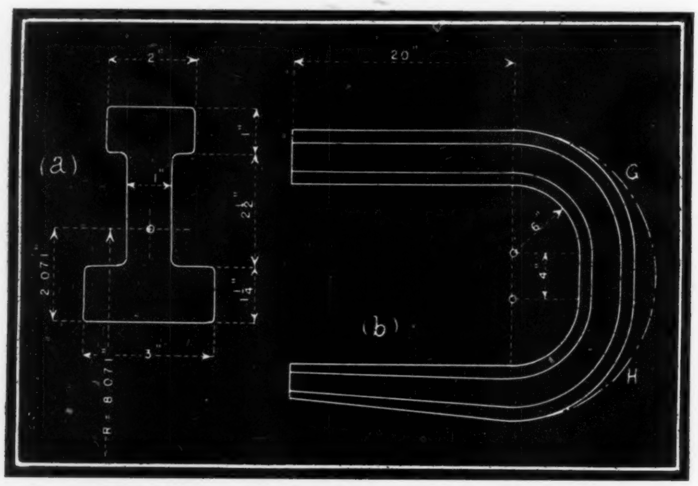


Fig. 10—In a yoke of this sort, deflection of the quarter bends accounts for over half the total deflection

relatively the stress which would occur across a section at X, *Fig.* 2; s_1 is the maximum stress according to the elementary Curved Beam theory (the exact value is a trifle greater), while s_0 would indicate the maximum stress concentrated at point Z, *Fig.* 2 because of the additional displacement of the neutral axis shown in the photograph *Fig.* 9. This emphasizes the importance of providing as large radii of curvature as possible in order to avoid such stress concentrations.

In spite of the fact that stress concentrations of this kind have been known and investigated for nearly a hundred years, design of machines is, in some cases, still made with an utter disregard for this condition. Use of many alloy steels does not permit of such plastic deformation at points of highly concentrated stress as would a softer, mild steel. The fact that alloy steels are more used today than ever before aggravates the condition arising out of the neglect of considering, in design, the displacement of the neutral axis of yokes under stress.

For example, let it be assumed that in *Fig.* 2, L is 6, $R = 4$, $h = 4$, the width 2, and the load 3000 pounds. The bending moment at X is 30,000 inch-pounds, but across the section at Z it is somewhat smaller due to the shorter lever arm. At X the maximum stress would be about 5600 pounds per square inch. At Z, based on the straight beam theory, the stress would be about 4800 pounds per square inch,

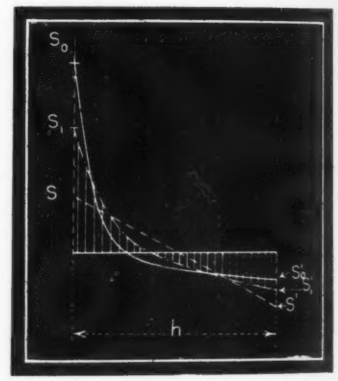


Fig. 11—Stress distribution over a section of depth, h; S, S_1, S_0 are tension

and according to the Curved Beam theory about 7400. Due to the displacement of the neutral axis still farther toward the inner edge, the maximum stress runs up to about 16,000 pounds per square inch, the calculation having been verified photoelastically.

Substitute Design for Safety Factor

In closing it should be mentioned that in the earlier days of machine design, unexpected failures dictated that a considerable allowance should be made beyond the calculated metal thicknesses, hence the "factor of safety." When more accurate stress analyses, especially by the photoelastic method, show high stress concentrations which formerly were not generally known or suspected, such stresses can almost be taken at their face value. Only a small margin need be allowed for "safety" if stresses alone are the determining factors in the construction. This safety margin is then allowed for the purpose of keeping the stresses below the elastic limit in rough portions of the machine part, such as roughness in a cast surface, or coarse machining ridges. Otherwise minute local failures will occur which, by repeated load applications, will develop into a serious fissure, and finally failure.

Selecting Special Motors

Part IX—Small Built-in Motors

By J. H. Staak
General Electric Co., Fort Wayne, Ind.

Fig. 1—Above—Shell-type motor parts showing comparison between straight-slotted and skewed rotors

Fig. 2—Below—Domestic food mixer utilizes vertical, series-wound motor for high power-weight ratio

DESPITE the fact that series motors were among the first of the fractional horsepower motors to come into existence, their general usefulness was not recognized until after the induction motor made its early appearance on washing machines. In the earlier days it was generally admitted that series motors could have no broad use because of their poor commutating characteristics and bad speed regulation and that the best outlook for them was their application to small fans and other relatively unimportant functions.

General interest stimulated by the possibilities of the wider use of power in many devices gradually brought the series motor to its present-day state of perfection and its enviable position of being able to serve at least 20 to 25 per cent of the known applications of fractional horsepower motors. Some of the previous drawbacks were turned into advantages and others were overcome by proper selection of applications to meet particular requirements. Today these motors take the major part in the motorization of office machines, household appliances, portable tools, and other miscellaneous portable devices as shown in *Fig.* 2.

Series motors are divided into two general types carrying a marked similarity in armatures and differing only in the construction of the field member; the straight series motor being recognized by its salient poles, interpolar gaps and its single form wound field coils, while the compensated motor is

known by its slotted stator, distributed field windings and its air gap extending over the complete periphery of the rotor. Visual comparisons may be obtained from *Fig.* 1.

The term "universal" as applying to series motors in general is a misnomer since these motors will not operate at like speeds between direct and 60-cycle current without providing some means to overcome reactance. They operate with a relatively weak field and an armature having a large number of turns to minimize the inductive reactance. To overcome this armature reaction a neutral commutating zone is provided in identically the same fashion as the effective brush shift in a compensated motor, to produce a neutral zone on the stator periphery.

Brushes are shifted against the direction of rotation until the coil undergoing commutation lies in a position of minimum or zero flux. In smaller frames the matter of proper brush positioning hinges somewhat on simplicity of housing design and the armature leads are carried around to the proper commutator bars to give an equivalent positioning of the commutated coil, thus keeping the brushes in a mechanically neutral position. The net results of employing means to overcome armature reaction are better power factor, efficiency and better commutation.

Several advantages accrue from the use of the compensated series structure. The distributed winding provides better flux distribution than can be secured with the single-coil salient-pole type; a greater amount of active material can be accommodated in the same diameter and stacking length since the inactive interpolar space of the straight series motor is converted into iron and copper; and better control can be exercised over the degree of compensation resulting from this, in improvement in speed regulation and better starting torques in favor of the compensated series motors in the larger frames only.

Advantages of Straight Series Motors

Both salient and compensated types of motors must, of course, be provided with proper ratios of field and armature ampere turns as well as keeping the armature turns per bar to a low value to obtain best commutating results by reducing the transformer electromotive force in the coils undergoing commutation when operating on alternating current sources of supply.

Simplicity of construction and ease of ventilation through the interpolar space mark the advantages of the straight series motor and the choice between the two types lies in the torque and performance requirements as well as in cost considerations.

The high speed of the universal and series motor, once considered a drawback, has been translated into a real advantage in that it permits light weight and large power outputs in comparatively

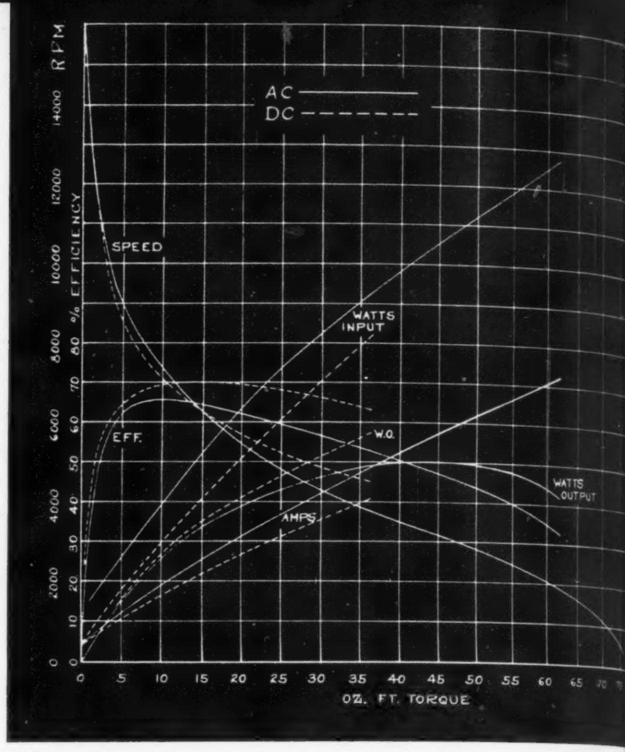


Fig. 3—Typical performance curves for 3/4-horsepower, 8000-r.p.m., 110-volt, alternating or direct-current, compensated, series motor

small dimensions so essential to many applications of a portable nature. Generally speaking, the salient-pole motor is used on ratings extending from 1000 to 1400 watts maximum output whereas the compensated motors are employed where maximum outputs are somewhat in excess of these values.

Have High Starting Torques

Speed variation because of load changes is common to all motors, except synchronous, but in the series motor it is considerably greater than in other motors since the torque is inversely proportional to the speed. The starting torques of these motors are high, ranging anywhere from two to three times the values of the alternating current full load torque which generally is chosen at a nominal speed of 10,000 to 12,000 revolutions per minute in salient-pole motors. Their no-load speeds reach values of 17,000 to 22,000, being limited only by brush and bearing friction windage and iron losses. The maximum output of salient-pole motors of moderate dimensions usually occurs at speeds of 4500 to 5500 revolutions per minute while that of the compensated motor is somewhat lower, falling into a range of 3000 to 4000 depending upon how heavily the frame size has been pushed.

General characteristics of compensated motors are similar to the salient-pole unit except that no load speeds rarely go beyond 16,000 to 18,000 revolutions per minute and that slip torque for the same values of maximum output are from 1⅓ to 2 times the value obtained in salient-pole motors. Motors of smaller dimensions than those used normally in portable tool or appliance applications may reach no load speeds of 25,000 to 28,000 revolutions per minute but are rarely used because of

their light outputs and the difficulty in supplying suitable bearings.

Flatter speed regulation curves can be obtained in compensated motors by providing them in a 4 to 6-pole construction but these motors are so critical of commutator setting that they are uncommon on devices other than pipe threaders and similar extremely heavy-duty applications. Typical curves of the salient-pole and the distributed-wound motors are shown in *Figs.* 3 and 4.

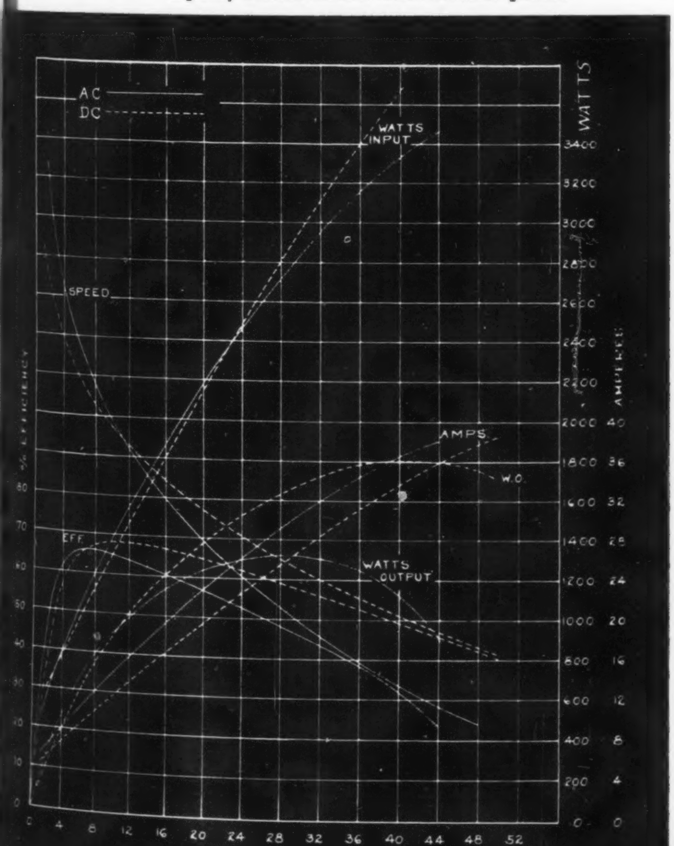
It is readily shown that as speeds are decreased the alternating and direct current curves have a rapid divergence. Although the direct current starting torques are not shown it can be estimated that they will be from 2 to 3 times the value of the 60-cycle torques. This indicates also that universal operation cannot be expected at the lower speeds but that it occurs at considerably lower points in the compensated unit. A comparison of the general characteristics shows that little is gained in efficiencies but that there is a marked improvement in speed regulation in the compensated motor.

Ratings Now Include One Horsepower

Normal ratings of salient-pole motors now extend to full load values of 1 horsepower whereas a rating of ½ to ¾ horsepower at 8000 to 10,000 revolutions per minute was formerly considered the limit for this construction.

Relatively speaking, series motors are inherently short-lived motors because of their high speeds and difficulties encountered in any commutating device. Their selection in application work must

Fig. 4—Performance curves for 3/4-horsepower, 10,000-r.p.m., series motor with salient poles



therefore be carefully weighed against the service they are expected to perform.

In the portable tool field are two general classifications, the first in which the motor is applied directly to the load and the second in which gears are built in to obtain proper speeds and torques. Both are definitely influenced by the size and weight without being restricted in power outputs necessary to perform the work expected. Typical samples of the first classification are such devices as routers, shapers and hand grinders. In the second group fall such applications as drills, screw drivers, hammers, sanders and edgers as shown in *Fig.* 5.

Considerations Affecting Selection

Other miscellaneous applications not falling directly into the portable tool class mentioned in the beginning of this article are such devices as industrial cleaners and blowers and insecticide sprayers, the success of which depends largely upon the high speed and light weight of the series motor.

Because of the relatively short life of series motors, their large outputs on tool devices and their variable speeds, the utmost care must be taken in the proper selection for each individual application. It is necessary in making an application of series motors to take into consideration such items as are discused in the following:

1. SERVICE FACTOR: Series motors are rarely used in applications where they may be called upon to operate many hours per day unless the power requirements are comparatively light and operating conditions are within reason. In general, most factory devices going into production lines are motored with constant-speed motors to avoid frequent servicing and replacement of brushes which limit the series motor to intermittent duty applications.

Constant-speed motors are utilized in portable tools where they perform heavy duties on production lines such as encountered in the building of automobiles. On these devices polyphase motors operate on higher frequencies to retain the characteristics of high speeds, large outputs, and light weights which characterize the series motor line.

2. DUTY CYCLE: The duty cycle is a measure of the temperature obtained in the driving unit and consequently determines to a large extent the size of motor that must be adapted to the job and the amount of ventilation necessary to permit it to operate at safe temperatures. In most series motors, constant case temperatures can be obtained in one hour or less on sustained loads, and if the normal cycle of operation equals or exceeds this time it must be classified as continuous operation.

3. VENTILATION: Small high speed motors, on account of their large outputs per pound of material, depend largely upon ventilation for successful performance. Because of the nature of the construction it is often found that housing tempera-

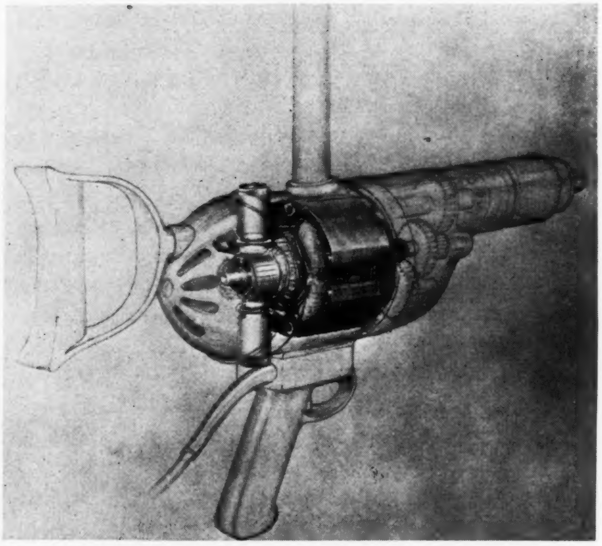


Fig. 5—Phantom view of a portable drill showing the application of high-speed shell-type motor to provide lightweight, compact drive

tures are exceedingly low, yet the motor parts may be on the verge of failure due to high internal coil temperatures.

Amount of ventilation and proper distribution in the frames which house the motor parts become major considerations in every series motor application. Most devices utilizing built-in motor parts—as in the case of the majority of portable units—mount the motor fields on lands or ribs to keep housings cool so that they will not be disagreeable to the touch. This provides an air space around the field core through which air can be drawn in addition to the interpolar spaces which are present in the conventional salient-pole series motors.

It is found that general motors applied to portable tools, domestic applications, and office devices require volumes of air ranging from 2 to 100 cubic feet per minute depending upon the losses which have to be dissipated. Fans must be of liberal size extending well beyond the armature diameter and frequently baffled to obtain best results. Machine designers often do not provide adequate ventilation. This frequently ruins an otherwise excellent design by failure to provide sufficient space for fans and baffles. Areas of ducts and their arrangements warrant considerable thought so that the maximum amount of power can safely be taken from motors or motor parts of small dimensions.

When considering, for example, a ¼-horsepower washing machine induction motor weighing 20 pounds against a ¼-horsepower series motor weighing 5 pounds with practically the same operating efficiencies the significance and importance of good ventilation is readily brought home.

4. Space Limitations: This must be known definitely to determine how much ventilation is required to keep the power unit cool.

5. Operating Speeds And Load Connections: Although these seem relatively unimportant they have considerable bearing upon temperatures and universal operation. First, it should be remembered that series motors must operate between 7500 and 10,000 revolutions per minute to obtain satisfactory universal characteristics and that the speeds are also important in the matter of power output and proper ventilation. If the load is connected through gears built into the motor housing, the heat generated through gear losses must also be handled by the motor fan.

6. Direction Of Rotation: In view of the fact that universal motors are given a definite off-neutral setting on the coils undergoing commutation and that this setting can be provided for only one direction of rotation, it is necessary to determine the direction of rotation for proper operation of the device. If reversible service is required it can be provided in a series motor by supplying a neutral setting at the expense of poorer commutation, shorter brush life, and lower efficiencies.

7. Starting Torque And Full Load Operation: Before any motor output or size selections can be made, it is necessary to determine the starting torques and full load requirements as well as any irregularities in such torque requirements during a cycle of operation. Peak torque loads often occur in devices which are operating mechanisms like adding machines, calculators and cash registers. Their sequence and time of occurrence during the cycle are vital to the proper motor selection for the device from the standpoint of allowable speed variations which may occur in such applications.

8. Shock Loading: One of the most difficult tasks that has been encountered in series motor applications is that of overcoming failures of armatures due to vibrating or shock loading imposed by the device. Typical examples are found in nut runners, screw drivers, concrete vibrators and hammers. Many schemes have been tried in the interest of eliminating the hazards encountered but only a few have met with any marked success.

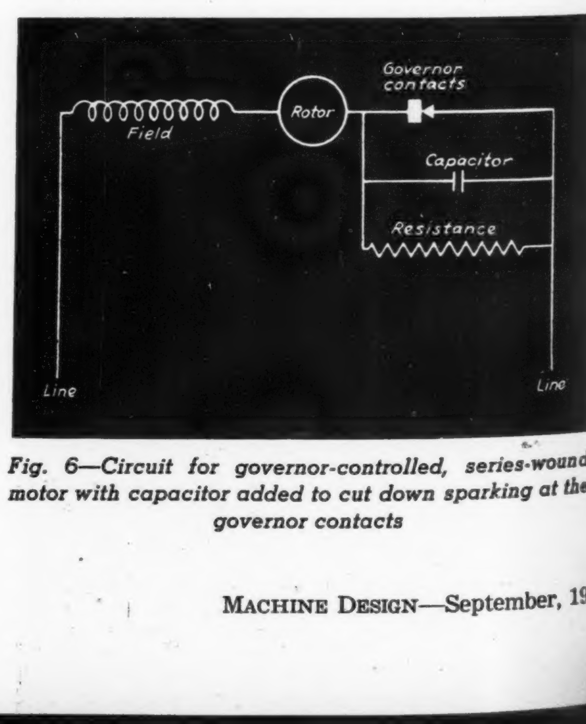


Fig. 6—Circuit for governor-controlled, series-wound motor with capacitor added to cut down sparking at the governor contacts

The most promising is a bar construction of reinforcing lead to the commutator which by far surpasses previous attempts of shaft stiffening, flexible lead constructions, and stranded lead reinforcing to the bars. It has been a boon to applications which have been a total failure with conventional methods of armature construction.

9. Acceleration Of Moving Parts: This is important in adding machines, calculating machines, typewriters, carriage returns and signalling devices, and must not be overlooked. It has a bearing on both torque and its relation to speed changes in the motor characteristics most suitable to meet the conditions desired. Many other items such as end-thrust requirements, allowable speed variations resulting from changes in voltage or loads, atmospheric conditions to be encountered, anticipated life, and facilities for servicing have an important bearing on the proper application of series motors. As a matter of fact the points for consideration are so numerous that every machine designer should avail himself of the application engineering service which almost all well known motor manufacturers supply, giving a complete analysis of the device and its operation so that the most efficient operation will result.

In many instances, modifications of the machine result in better fitting of motor characteristics and economies which otherwise may have been overlooked.

Domestic Applications Are Built-in

In view of the fact that the foregoing discussion deals mainly in the motorization of devices which are typical of high-speed, heavy-duty built-in sets of motor parts, let us turn for a moment to domestic applications typified by such devices as the kitchen mixer, vacuum cleaner, etc. General characteristics of motors applied to these devices are similar to those used in tools and portable devices in that they operate at high speeds, yet it is unnecessary to build them for heavy duty since they are not subjected to the abuse and long life so important to the portable tool and device field. They could readily be classified as a light-duty application as distinguished from the motor parts application in tools. It has been customary in the past that manufacturers of the above-mentioned devices utilize only armatures, fields and brush mechanisms and build them in as an integral part, while the experience with office devices has been the exact opposite. This is probably due to the fact that much more pleasing designs can be obtained by building in, while the nature of office applications does not require such treatment because of the fact that decorative covers for other mechanisms might as well be utilized to cover the power unit also.

Prior to the advent of the contact-governed motor in the office appliance field, such schemes as

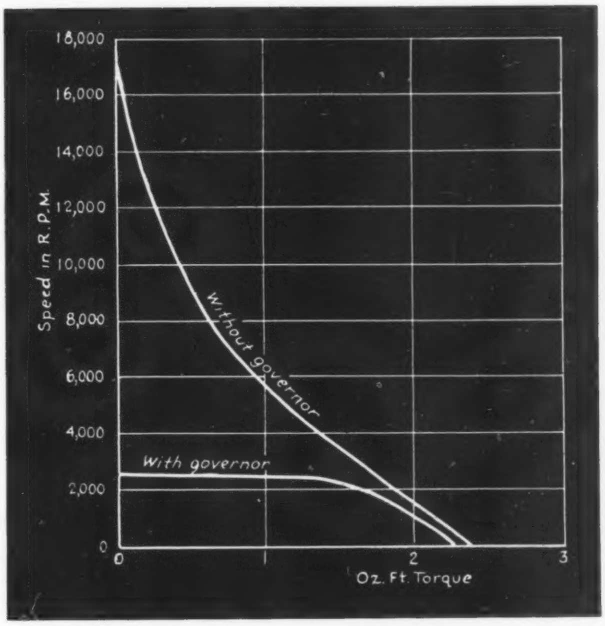


Fig. 7—Speed-torque curves for a series-wound motor with and without governor

shunted-series motors, dashpots, and mechanical regulators, were looked upon as the answer to the various problems but these met with only mediocre results.

Centrifugally operated contact mechanisms controlling electrical circuits were used in a few highly specialized fields for more than twenty-five years but they were entirely too costly to apply to office machines. They served mainly such devices as communication machines and temperature recorders.

More general fields for such governors were not developed until about 15 years ago when the governor-controlled series-wound universal motors were completed for use on adding and calculating machines and met with immediate acceptance by their users.

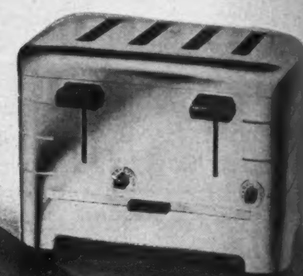
Governors Increase Use

Today by far the greatest percentage of adding and calculating machines in the United States use fundamentally the same governor-controlled motors. While there are certain unfavorable features limiting the field of application of contact-governed alternating current and direct current motors they have done a creditable job and have proved themselves satisfactory for the purpose intended.

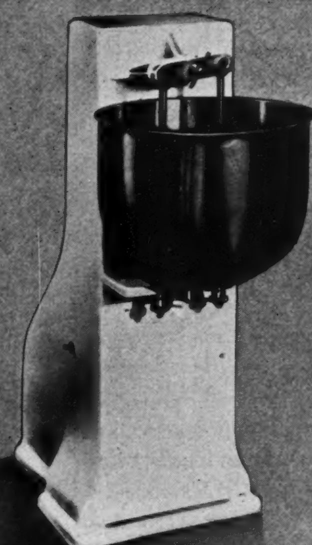
Governors when applied to series motors are connected in series with the windings to make and break the motor circuit for speed control. There are two general types of governors, a center contact governor which obtains a wiping action at the contacts, and the type in which contacts rotate with the shaft and give a fanning action by their rotation. For suitable applications the later general

(*Continued on Page* 130)

Below—Automatic temperature-controlled t- ing mechanism assures uniformity in Toastm- toaster. Manually set timer control is co- pensated by thermostat as the unit heats, the- by eliminating preheat time. Use of expo- contacts is avoided by utilizing mercury switch- es to control power in each of the two separ- units. Traversing of bread slices past zig-z- guard wires during operation eliminates po- bility of uneven toasting



Above—Spindle, mounted in precision preloaded ball bearings, driven through a single V-belt running over 4-step sheaves, provides a four-speed drive for Landis threading machine varying from 270 to 845 revolutions per minute. Completely enclosed spring-loaded toggle mechanism effects gripping of the work, eliminating collets



Right—Built without glands or seals other than the clearance between working parts, the Logeman homogenizer utilizes no lubricant for pistons or valves except that provided by the fluid being processed. A piston, mounted on the end of each articulated rod driven by a rocker, fits into an extension of the valve housing in the bottom of the kettle. Reciprocation of pistons permits the fluid to flow alternately into each chamber

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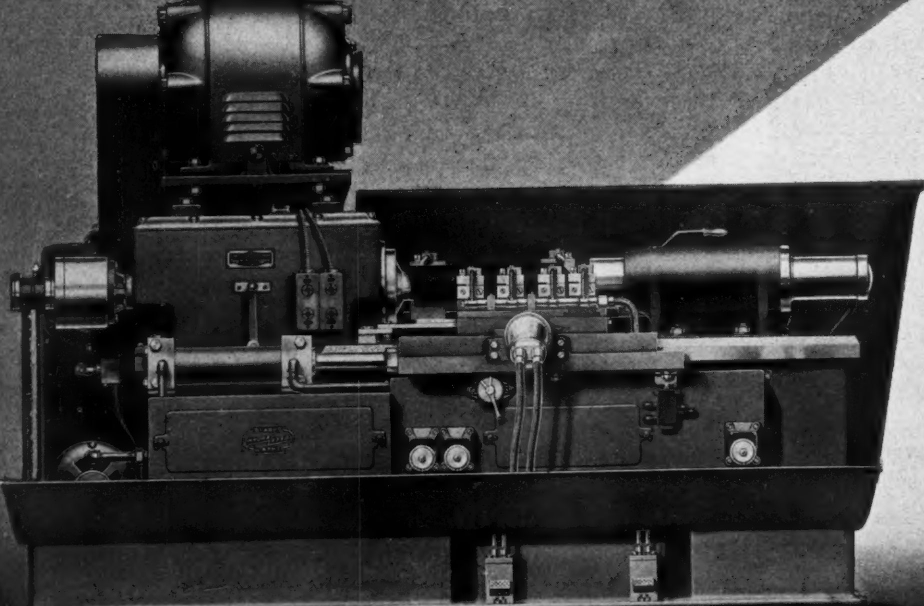
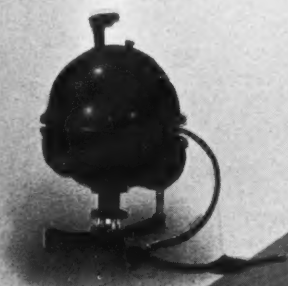
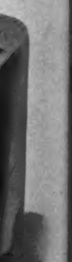
Below—Designed for use of tungsten-carbide tools, triple-geared, 4-speed headstock of Sparks shell lathe is adaptable for finish turning 75, 105 and 155-millimeter, as well as 8-inch shell, heavy-duty roller bearings are used on all shafts and spindles. Synchronization of headstock with stop movements of tools mounted on carriage is obtained by a hydraulically operated, multiple-disk clutch and brake



Above—Elimination of ... polished finish on all par- minimizes the risk of conta- pendence upon throats fo- guide sleeves of stainless throughout practically its e- rating a seating angle of ... ating pressures, utilizing ... trusion thro-

elow—Completely diffused light for checking olors with Comparascope is obtained by use a 10-inch diameter hollow aluminum sphere. nterior of sphere is specially coated to provide dequate lighting with two lamps mounted on e outer surface. Tubular, monocular and nsless eyepiece focuses below the port at ottom of vertical axis of sphere, permitting ccurate color comparison of different materials without regard to their surface finish





Above—Pneumatic controls effect sequence regulation and also provide power for turret rotation in American pressing machine. A cam on underside of turret operates air valve. Admission of air to cylinder preloads a heavy compression spring through an intermediate linkage mechanism. Then operation of trigger permits release of spring energy to rotate the turret. Mounted on roller bearings, turret is special cast iron capable of taking a high polish

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Left—Incorporating a special electronic circuit operating on a "currentless contact" principle, displacements of a minute fraction of an inch are detectable by Carson rubber micrometer. A green signal light built into the instrument flashes the instant contact is made with the material whose thickness is being measured

Below—Perfect alignment of shaft bearings is maintained by heavy nickel cast iron chain race in Diehl veneer splicer. Maintenance of fine adjustment and elimination of distortion is provided by I-section C-yokes adequately cross braced. Thermostatically controlled electric heaters, mounted above and below stock adjacent to floating roller chains, effect continuous splicing or welding of veneer sheets without abrading

he application of a highly uid facilitates cleaning and erry-Burrell viscolizer. De- unger is avoided by use of e accurate plunger control plit-flow plug valve incorpo- akes possible reduced oper- ciple as well as that of ex- high velocities

MACHINE

Editorial

DESIGN

Twelfth Anniversary

WITH this issue MACHINE DESIGN celebrates its twelfth anniversary. Would that the conditions of peace and prosperity existed today that obtained at the time of the magazine's inception in September, 1929. Let's fervently hope it won't be long before peace and more logical and lasting prosperity are back again.

During the twelve years since it was published this journal has seen—and fostered—many changes in the design field. Too numerous to discuss in detail, one specific example might be cited as the elimination of the gearshift lever on automobiles which, though not finally completed, is well on the way; another, more general, example is the "cleanlining" of machines throughout the design field. One would not recognize the machines of ten years ago compared to those of today.

But little was it thought when the magazine was started that it would run into a period of conflict between nations. If a war were to occur, however, it was generally conceded that it would be a battle of machines. This proved only too well to be the case, and MACHINE DESIGN is proud of the part it has played and is playing in the development of mechanical equipment for defense that is second to none.

MACHINE DESIGN has always thoroughly enjoyed its association with its field and its readers, and aspires to the hope that it has filled a distinct need as well as serving a clearly defined purpose. Evidence exists, as indicated in the foregoing, to prove that such aims have been fulfilled. It will be our constant purpose during coming years to continue to maintain the high standard of quality that might rightly be expected of a technical journal devoted in its entirety to the machine design field.

Graphical Calculation of Helical Springs

By Carl P. Nachod
Vice President
Nachod & United States Signal Co.

Rectangular nomògram for the calculation of steel compression springs which appears on the following page is based on the Reuleaux formulas

$$P = \frac{\pi D^3 S_1}{8D} \qquad f = \frac{\pi D^2 S_1}{Gd}$$

where P = load, d = wire diameter, D = mean spring diameter, f = deflection per coil, G = torsional modulus of elasticity and S_1 = fiber stress.

The actual fiber stress obtained from the chart is $S = S_1 y$ in which y is the Wahl factor which varies from 1.15 to 1.4 for values of the spring index, $r = D/d$, from 10 to 4. The approximation of the Wahl factor used in the chart is $y = (r+1)/(r-.4)$ for the range indicated.

For $r = 8$, spring proportions are good. Decreasing r gives long thin springs; increasing it, shorter and larger diameter springs. The chart shows the actual stress due to the curving of the wire, which is higher than that given by the Reuleaux formulas. The torsional elastic modulus G for steel has been taken as 11½ million.

Most effective use of the chart is obtained by drawing a pair of 90-degree "cross hairs" on a transparent material. Superimpose this templet on one of the two squares of the chart so that three of the lines intersect the proper values on the respective scales. The fourth value can be read under the fourth line.

Thus the dashed lines in the right-hand square show that a helical steel compression spring wound with .09-inch diameter wire, a spring index of 8 and stressed to 60,000 pounds per square inch will exert a force of 20.1 pounds.

Applying the templet to the left square over the same values of P, r and d, the compression per turn is found to be .0795-inch.

Having determined these values for a particular spring, other values may easily be calculated:

Mean Spring Diameter $= D = rd =$ .72-inch

Outside and Inside Spring Diameters, respectively $= D \pm d =$ .81 and .63-inch

Total Compression for 25 active turns $(n) = F = fn = 2$ inches

Spring Rate $= R = P/fn = 10$ pounds per inch

Allowing 10 per cent of d for space between turns,

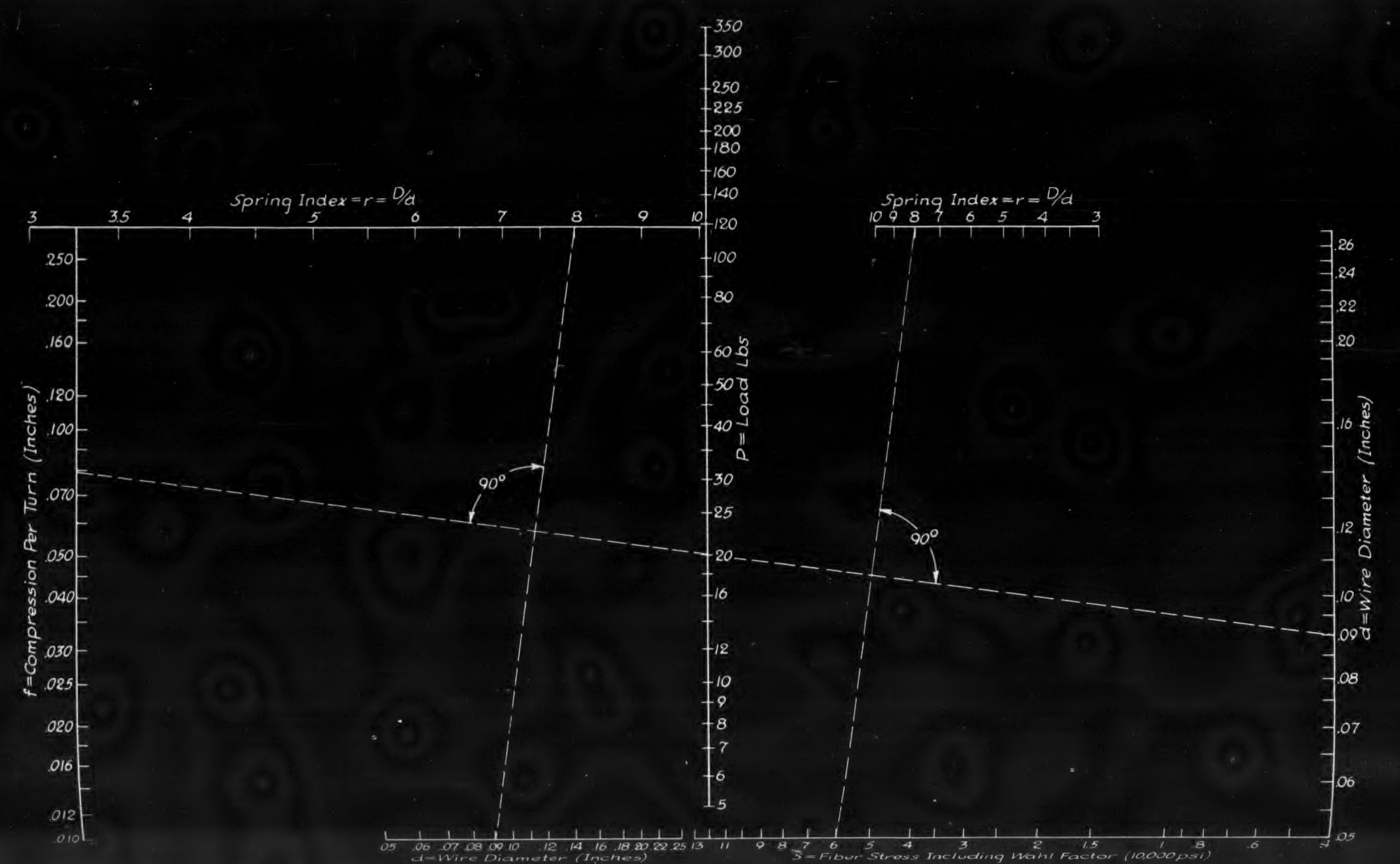
Compressed Length $= l = 1.1d(n+m) = 2.7$ inches

where m is the number of dead turns at the ends, computed by

$$m = 2.5 - (5/n) = 2.3$$

Free Length $= L = l + F = 4.7$ inches.

If the assumed quantities are P, r and f, then use the left square to find d, and the right square to find S (the maximum fibre stress) which should not exceed a safe value. For many operations, as with auto valve springs, this depends on the range of stresses. See Engineering Research *Bulletin* No. 26, University of Michigan, "Permissible Stress Range for Small Helical Springs" by F. P. Zimmerli.



Spring Index = r = D/d
Spring Index = r = D/d
f=Compression Per Turn (Inches)
P = Load Lbs
d=Wire Diameter (Inches)
d=Wire Diameter (Inches)
S = Fiber Stress Including Wahl Factor (10,000 psi)
90°
90°

Professional Viewpoints

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

". . . could fill the need"

To the Editor:

Your editorial in the August issue regarding a "Flying Squad" of tool designers interested us. We are just getting started in the designing business and perhaps could fill the need for such an organization.

We are a group of young men, thoroughly experienced in the design of jigs, fixtures, gages and special machinery. One of our number has had considerable experience as a die-casting designer, not only of parts but of dies and tools as well.

If you receive other responses to your editorial, especially from people interested in obtaining the services of a "Flying Squad", we would certainly appreciate your forwarding their inquiry to us so that we might contact them with a view towards solving their design problems.

—WILLARD F. BUB,
Acme Engineering Co.

". . . points should be clarified"

To the Editor:

Referring to the article by Messrs. Owen and Kirkish in your June issue on "Selecting Special Motors for Frequent Starting and Reversing", much useful information is presented by the authors. In view of the importance of this subject, especially at the present time when so many new types of special-purpose machines are being designed, a few points which confused the present writer should be clarified by constructive comment.

In leading up to Equation 2, the authors state "It cannot be stressed too strongly that accelerating a mass of fixed inertia to a specified speed will result in a fixed value of rotor heating." This statement would apply only to the repeated accelerations of the same or identical motors. A motor of 90 per cent efficiency would have twice the heat dissipation of a motor of 95 per cent efficiency, although both motors accelerate the same inertia to the same speed.

In Equation 5, the motor r.p.m. should of course be squared, so that the right side would read "91.2 WV^2 / N^2. It is assumed that a typographical error is responsible for this discrepancy.

Under TABLE I for a motor driving a high external inertia load, the "Rotor $WR^2 = 70$ lb.-ft.^{2}" was evidently intended to read 0.70 lb.-ft.^{2} to be in line with the stated external inertias.

Equation 6 is called into question. The premise "for any rotor copper loss $I^2_2r_2$ there results a corresponding stator copper loss $C_1I^2_2r_1$" is correct, but is no basis for Equation 6. The term "stator accelerating loss" appears self-contradictory, since the acceleration is useful external work, and the internal copper losses vary widely with the motor efficiency for any given acceleration task. Basically, Equation 6 assumes that Equation 1 expresses the actual heat dissipation or copper losses during the acceleration period.

Equation 11 is correct, provided the actual joules heat loss is substituted for "$J_{(total)}$" which is considered an irrelevant quantity as far as the cooling requirements of the motor are concerned.

Equation 12 is theoretically incorrect and apt to give more approximations if applied to actual cases where the torque varies over a considerable range during the acceleration. This equation is true only when the torque is constant or when the time-average torque is effective at the speed $N/2$, which is an arbitrary assumption. Equation 12 further assumes that work equals torque × angular speed, which gives *power* not work.

Actually, the stored kinetic energy must be rigorously expressed as follows:

Stored rotation $K.E. = \int_0^\theta T d\theta$, where T is the momentarily varying torque and θ is the total angle in radians traversed during the acceleration; and

Stored translation $K.E. = \int_0^s F ds$, where F is the momentarily varying force and s is the total distance travelled during the acceleration.

Stated in words, the stored rotational kinetic energy is equal to the area of an angular-work

(Concluded on Page 138)

Applications

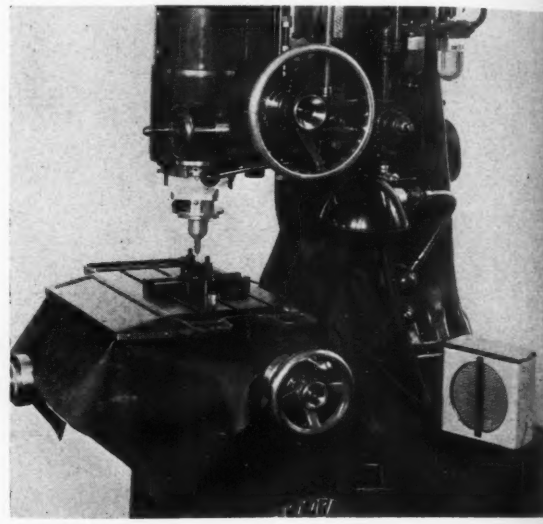
of Engineering Parts and Materials

Detects Contact with Work

GRINDING small holes with diamond-impregnated laps makes it difficult for the operator to determine when the lap is in contact with the wall of the hole as well as how much metal is being removed. With a diamond lap no sparks are emitted and, since the hole as well as the lap spindle is small, visibility is limited.

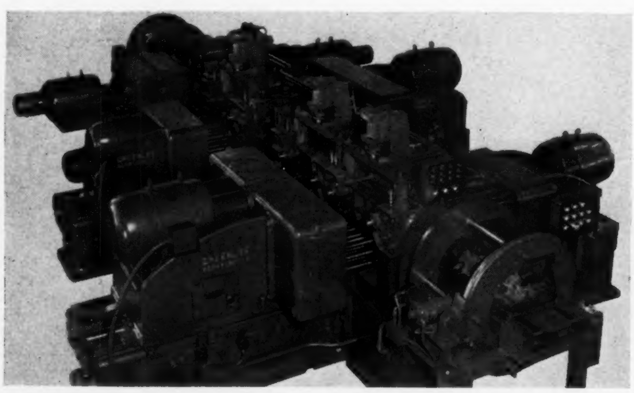
Trouble of this nature has been eliminated in the Moore jig grinder, the design of which is discussed in an article in this issue, by use of a vacuum tube amplifier and loud speaker. Capable of being adjusted to any degree of sound amplitude, the instrument is connected to the work. The operator can thus detect aurally when the lap contacts the work and, with a little experience, estimate the rate of stock removal.

Elimination of destructive abrasive grinding dust on cross-slide ways and micrometer screws is provided by neoprene aprons. Various types of sliding and telescoping guards were investigated before this solution was adopted. The aprons used are unaffected by oil and may be cleaned easily.



Built-in Motors Aid Control

COMPRISING virtually a complete production line built into one machine, the Greenlee precision multiple-drilling machine consists of three separate, two-ended drilling units. Powered throughout by Howell motors driving the six individual spindle transmissions, precise speed control is attained. Supplementing manual controls, snap-action limit switches which are mounted on the base and operated by the motor carriage provide automatic sequence control for a given cycle of operations.



Plastic Facilitates Study

INVESTIGATION of the performance of the working parts of fully enclosed mechanisms usually offers the engineer serious difficulties. The problem was solved in the case of Trident water meters by building the model illustrated.

Housings and fittings are machined from solid blocks of Plexiglas. Use of this optically clear acrylic plastic permits unobstructed vision of the operation of the gears and bearings which are of conventional metal construction. Being highly resistant to water and the effects of exposure, the plastic maintains its clarity indefinitely.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AIRCRAFT ★ AUTOMOTIVE ★ INDUSTRIAL

ASSETS *to a* BOOKCASE

Prevention of the Failure of Metals Under Repeated Stress

By the staff of the Battelle Memorial Institute under the auspices of the National Research Council of the National Academy of Sciences; published by John Wiley & Sons, Inc.; 6 by 9 inches, 273 pages, clothbound; available through MACHINE DESIGN *for* $2.75 *postpaid.*

A handbook prepared for the Bureau of Aeronautics, Navy department, this book discusses fatigue and the prevention of failure of parts. Lack of knowledge or appreciation of engineering principles among designers and others having to do with the design and construction of aircraft has been responsible for many fatigue failures. This book, therefore, presents much needed information on the subject.

Included are illustrations of characteristic failures, drawings, curves and tabular data useful in designing machine parts. Discussions cover effects of surface scratches, threads, fillets, keyways, splines, collars, holes, corrosion, inclusions, surface coatings, grain, embrittlement, harmonic vibration, etc.

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Electromagnetic Devices

By Herbert C. Roters, Director of Research, Fairchild Aviation Corp.; published by John Wiley & Sons, Inc.; 6 by 9 inches, 561 pages, clothbound; available through MACHINE DESIGN *for* $6 *postpaid.*

Engineers concerned with the design and application of magnetic devices will find this book helpful. Divided in two parts, it develops in the first the fundamental background theory and methods applicable to magnetic circuits and nonrotary electromagnetic devices. The second covers their application to a variety of problems.

These solutions are developed in general terms, each followed by a numerical solution for a particular problem. Special methods are developed to handle the calculation of magnetic leakage which has always been difficult of accurate solution. Emphasis is placed on graphical methods for practical problems involving nonlinearity caused by hysteresis and saturation in iron and the variation of magnetic leakage with motion.

Quantitative data are presented, both on magnetic materials and electromagnets, many of which are original. New theory is developed for estimating permeance as well as for magnet design. Also covered in the book are magnetic properties of alloys, calculations of circuits, design of coils, time-delayed magnets, high-speed magnets, alternating-current magnets, and relays.

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Mathematics for Engineers

By Raymond W. Dull, consulting engineer; published by the McGraw-Hill Book Co., New York; 8 by 5½ inches, 780 pages, clothbound; available through MACHINE DESIGN *for* $5 *postpaid.*

This second edition, with minor changes, is a quick, convenient and reliable reference book for engineers who wish to review or extend their information on those phases of mathematics important in engineering work. Material is presented in the form in which it is needed, covering simple numerical computations through integration and summation. Slide rule computation is given special treatment as are absolute and relative errors.

The book is helpful to those engineers who feel the need of a source of reference, who have grown somewhat rusty in their mathematics or who need a text for study. Because the book is presented as a review, the various branches of the science are treated as interrelated subjects.

□ □ □

Patent Fundamentals

By Leon H. Amdur, member New York Bar, former patent examiner; published by The Chemical Publishing Co., Inc.; 305 pages, 5½ by 8½ inches, clothbound; available through MACHINE DESIGN *for* $4 *postpaid.*

An easy reading discussion of patent law and its protection, this elementary yet thorough book gives the reader a workable knowledge of patent system and tests for patentable inventions. Included are preparation and prosecution of a patent with reproduction of actual memoranda and papers. Enlightening discussions of function and drafting of patent claims, patent searches, and classification of patents are helpful.

Engineering or Purchasing for Defense?

Take this SINGLE Step to a TRIPLE Saving!

***Call in P-K!* Get together with a Parker-Kalon Assembly Engineer to gain man-hours, speed work**

American Gas Machine Company speeded up operations 25% by eliminating tapping in joining this burner assembly to the manifold casting. They simply jig-drill 3 holes and with a power tool speedily drive in Parker-Kalon Self-tapping Screws for secure fastenings.

Your production won't be slowed-up by "doubtful screws" . . . screws that look all right but some of which fail to work right . . . when you use genuine Parker-Kalon Self-tapping Screws. For a further assurance of savings in hours and dollars is the Parker-Kalon Laboratory – without counterpart in the industry – which maintains a rigid quality-control over all Parker-Kalon products.

In daily contact with engineering, production and purchasing departments of plants like yours, Parker-Kalon Assembly Engineers are thoroughly familiar with present-day assembly practices. In 7 out of 10 cases they have pointed the way to *a triple saving* through (1) simplifying assembly work, (2) saving operations, (3) eliminating a common cause of production "slow-ups". These advantages are made possible by the properly engineered use of Parker-Kalon Self-tapping Screws.

Whether your assemblies are thin or heavy sheet metal, castings or plastics, you can profit by this simpler fastening method, which eliminates tapping and all its problems; the fumbling that goes with bolts and nuts and the difficulties of riveting in hard-to-get-at places; the need for lock washers to make secure fastenings or tapping plates on thin sheet metal and inserts in molded plastics. You require no major production changes nor special tools to use Self-tapping Screws.

Call for a Parker-Kalon Assembly Engineer. His recommendations will be unbiased because Parker-Kalon makes all types of Self-tapping Screws–thread forming and thread cutting – for every assembly of metals and plastics. Or send your assembly details by mail for recommendations and samples. Parker-Kalon Corp., 192-200 Varick St., New York, N. Y.

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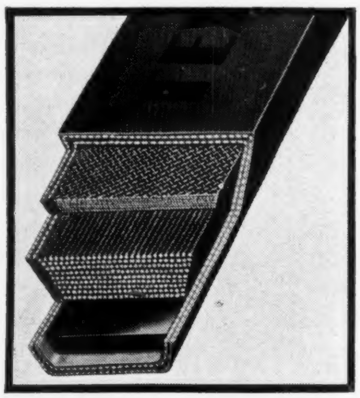
New PARTS AND MATERIALS

V-Belts of New Design

IMPROVEMENTS have been made in V-belt structure by Allis-Chalmers Mfg. Co., Milwaukee. All its Texrope V-belts are now of the new "Super 7" laminated design, based on the Vogt formula, to include more strength and flexibility, better service and longer life. Smaller cords in the new belt permit use of more cords per belt with a re-



sulting greater strength and less stretch. Each cord is individually embedded in heat dissipating rubber to reduce internal belt degeneration. Made in matched sets to assure uniform, smooth running, highly efficient drives, the belt has a live rubber bottom cushion which absorbs impact of operation. The central cord portion transmits power at the effective pitch diameter. Bias cut fabric prevents "dishing" and assures transverse stability. A two-ply rubber-impregnated fabric cover prevents destructive agents from reaching the vital belt elements, resulting in a high-grip coefficient between belt and sheave walls.

Breaker Requires No Fuses

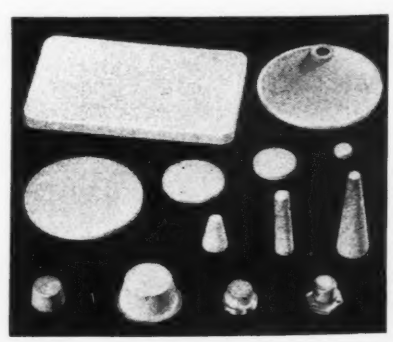
AVAILABLE in two types, M-1 and M-2, a new multi-breaker is announced by Westinghouse Electric & Mfg. Co., East Pittsburgh, for use on any normal duty application that otherwise would require fuses or fused switches. These 15 to 100-ampere, 2 or 3-pole breakers can be used on alternating current circuits up to 230 volts. The sheet steel enclosure is dust-resisting with 10 concentric knock-outs or conduit or cable connections.

The bimetal thermal element is actuated by overload or short circuit and causes the breaker to trip. While the breaker trips immediately on short circuit or dangerous overload, an inverse-time characteristic allows it to remain closed during tem-

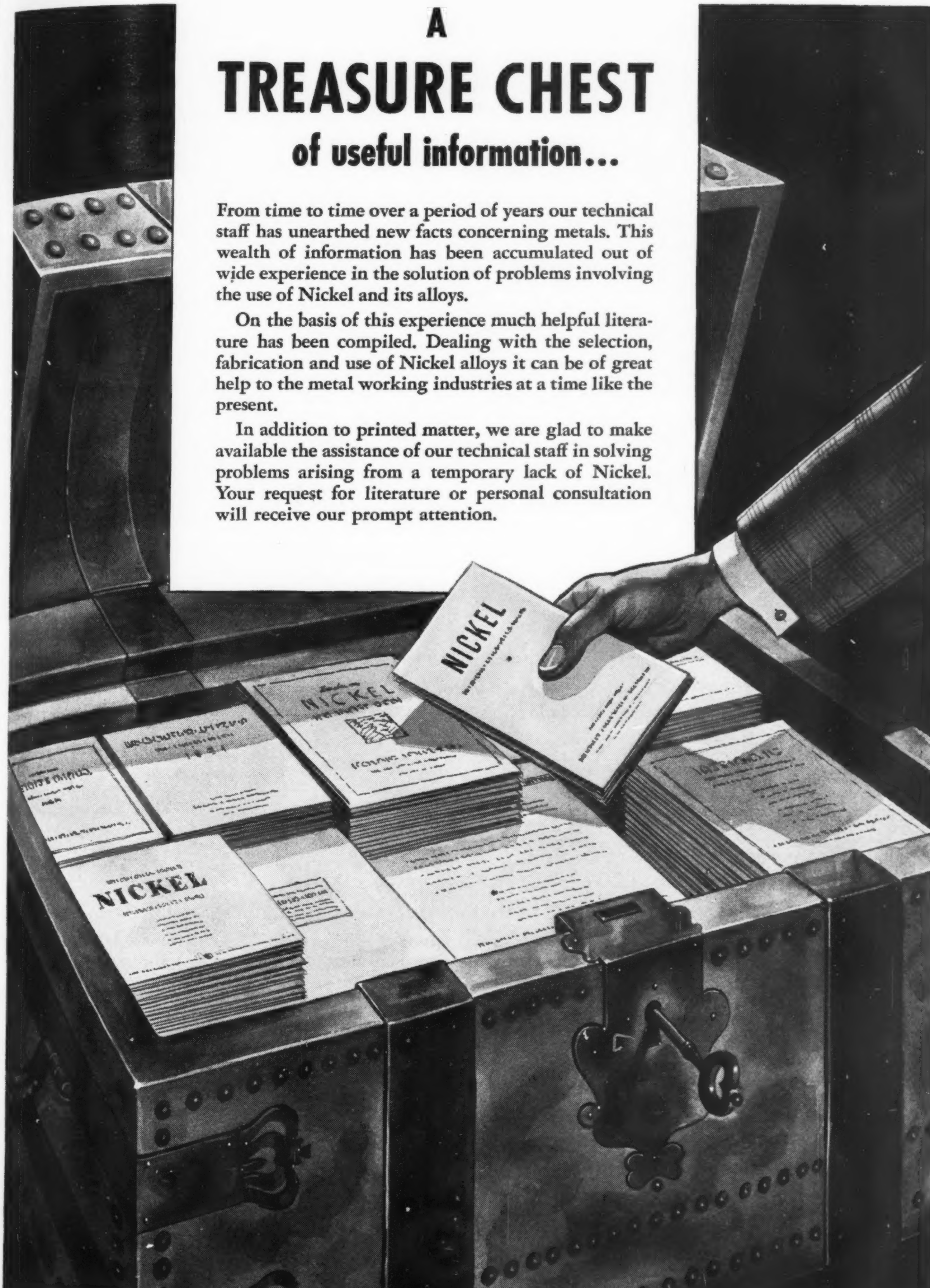
porary harmless overloads. An indicating target on the enclosure cover shows when multi-breaker has tripped.

Porous Metal for Filtration

POREX, a porous metal recently introduced by Moraine Products Division, General Motors Corp., Dayton, O., has been developed to remove unwanted materials from fluids and to alter flow of fluids to meet specific requirements. These basic functions have unlimited variations in many products and processes such as pumps, refrigerators,



fuel lines, lubricating systems, filtering, diffusing, screening, separating and mixing a wide range of fluids and gases. Its physical characteristics—chemical composition, structure, porosity, strength, ductility—and its shape and size can be varied



A TREASURE CHEST

of useful information...

From time to time over a period of years our technical staff has unearthed new facts concerning metals. This wealth of information has been accumulated out of wide experience in the solution of problems involving the use of Nickel and its alloys.

On the basis of this experience much helpful literature has been compiled. Dealing with the selection, fabrication and use of Nickel alloys it can be of great help to the metal working industries at a time like the present.

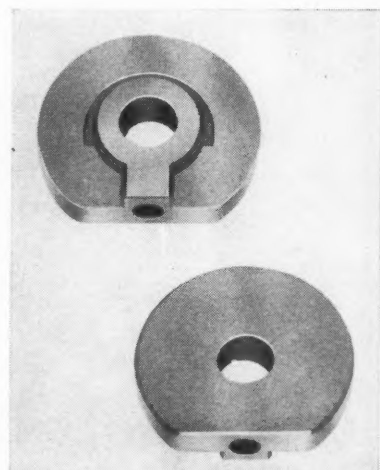
In addition to printed matter, we are glad to make available the assistance of our technical staff in solving problems arising from a temporary lack of Nickel. Your request for literature or personal consultation will receive our prompt attention.

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK, N. Y.

within limits to suit specific applications. Porex can also be bonded to steel or copper. Tensile strength is up to 10,000 pounds per square inch, depending on porosity which also can be varied. Ductility ranges between 3 and 15 per cent in tension, and considerably higher in compression. It is not readily machined or ground, and is furnished in disks, sheets, cylinders and truncated cones. Special shapes such as flanged cones, plugs, parts with multiple diameters and sections with surface configurations of varying depths and contours are also obtainable.

Expand Line of Porous Bearings

TO ITS line of Selflube porous bronze bearings, Keystone Carbon Co., 1935 State street, Saint Marys, Pa., has added several new shapes and sizes. These bearings, made from powdered alloys, are molded to size in shape desired, baked and finally quenched in oil. Average porosity of 35 per cent enables the bearings to store a large amount of oil,



forming a protective, continuous oil film on the surface of the bearing. While additional lubrication is recommended for heavy-duty or continuous operating units, in many cases the oil reserve lasts the entire life of the application, eliminating use of oil vents or grease cups. Because of the high tensile strength, 35,000 pounds per square inch, the bearings will carry maximum loads without distorting or breaking. Low friction coefficient and the self-lubricating qualities prevent excessive temperature, speed reduction, noise and scoring of shaft.

Breakers for Power Control

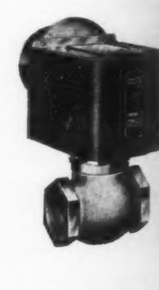
INDUSTRIAL multi-breakers for light and power control are being produced by Colt's Patent Fire Arms Mfg. Co., Hartford, Conn. Known as Types M1 and M2, these breakers can be supplied with either two or three poles and in ampere ratings from 15 to 100, for 230 volts alternating current systems only. Type M1 indicates ampere ratings

of 15 to 50; Type M2, ampere ratings of 50 to 100. The multi-breakers provide automatic protection against short circuits and overloads, and permit resumption of service by movement of the operating handle after cause of interruption has been removed. Common trip assures the opening of

all poles when current is interrupted on any one pole. Tripped position of the breaker is indicated through small window in front cover. The breakers are furnished with both thermal and magnetic trip—the thermal trip providing a time delay on small overloads, while the magnetic trip assures instant operation on dangerous short circuits or high overloads.

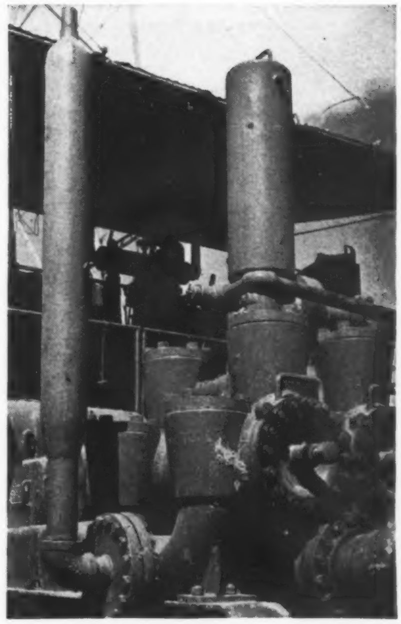
Control Valve Is Pneumatic

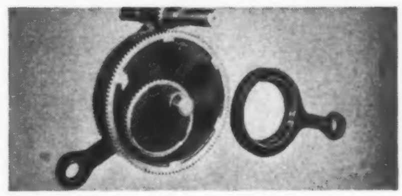
Many new features are incorporated in the pneumatically operated control valve announced by Minneapolis-Honeywell Regulator Co., 2950 Fourth avenue, South Minneapolis. Movement of the valve disk is accomplished by means of a scissors type linkage permitting more accurate control near the closed position of valve where accurate control is most needed. A built-in "Gradutrol Relay" provides maximum power to change position of the valve disk even for very slight changes in demand by the controller. Specially constructed bellows of molded neoprene are used in the power unit and assure long operating life with maximum power in a minimum of space. An external indicator shows the position of the valve disk. Provided also are adjustments for maximum and minimum flow and for throttling range. The valve body is single seated, bronze with brass trim, in sizes from



IN THE NEWS
WITH BANTAM BEARINGS

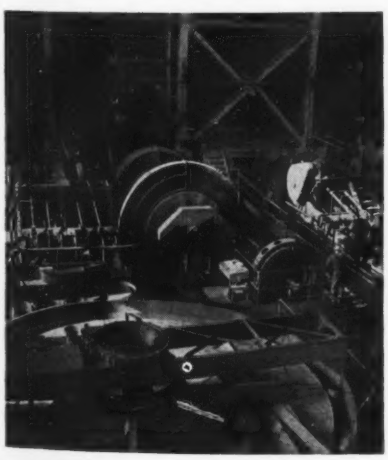
WATCHER OF THE SKIES in the nation's defense, the first of the new Bofors 40-millimeter mobile anti-aircraft guns rolled out of the plant of The Firestone Tire & Rubber Company weeks ahead of schedule, with General Charles T. Harris and John W. Thomas, president of Firestone, at the controls. For this newest defense weapon, Bantam supplies special double-direction thrust bearings on which the gun units turn—another instance of Bantam's skill in the manufacture of bearings for unique engineering requirements.



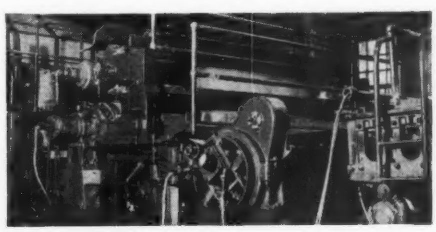


OIL FIELD PUMPS ARE WORKING day and night to keep pace with the constantly growing demand for petroleum products. Typical of the progressive design that enables the pumps to withstand twenty-four hour heavy-duty service is the 7½ x 14 power pump built by Emsco Derrick & Equipment Company. A specially designed Bantam Straight Radial Roller Bearing, 20″ I.D. x 25″ O.D. x 3″ wide is used on the eccentric.

BANTAM'S ENGINEERING SKILL and cooperation with machine builders give added assurance of satisfactory bearing performance. Bantam engineers, with wide experience in the performance of every major type of anti-friction bearing, will select or design the bearing that best meets your needs. For *better* machines, more and more manufacturers TURN TO BANTAM for their bearing requirements.



FINE GRINDING is an extremely exacting job in this cement mill installation of equipment built by The Dorr Company, Incorporated. This particular unit is used to convert 1-inch stone into fine powder. Other Dorr Classifiers are playing an important part in preparing ores for treatment by metallurgical processes. On many of the Dorr Classifiers—particularly on larger models for heavy duty—Bantam Ball Bearings are now standard for the eccentric strap.



IN HIGH-SPEED PAPER MACHINERY, Bantam Bearings of many types find wide application. Bantam Mill Type Bearings are used in this suction press roll built by Beloit Iron Works for a leading paper mill. Bearings of this type are also extensively used in steel mill and rubber mill machinery.

BANTAM BEARINGS

STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

BANTAM BEARINGS CORPORATION · SOUTH BEND · INDIANA

½ to 3 inches for screwed connections. The characterized V-port guide may be adjusted for lift up to ¾-inch maximum.

Solderless Lug Available

SET screw type of wire connecting lug, made of seamless, pure electrolytic copper with a heavy brass check-proof, heat reducing shell, has been announced by Ideal Commutator Dresser Co., 1059 Park avenue, Sycamore, Ill. The new solderless lug has full current-carrying capacity evenly distributed from wire to lug. Eight sizes are available for No. 14 wire to 2,000,000 circular mils cable. Each size is satisfactory for a wide range of wire. One or two-hole types can also be obtained with square or round ends. Special size and angular lugs can also be furnished.

Terminal Strips Are Compact

IN ORDER to carry the required current safely, compact terminal strips with maximum metal-to-metal spacing have been designed by Howard B. Jones, 2300 Wabansia avenue, Chicago. The

body is of heavy molded bakelite, with barriers between each set of terminals, following around the edge of the strip and terminating with the base. The new 150 series, comprising three sizes, 150, 151 and 152, are added to the 140 series.

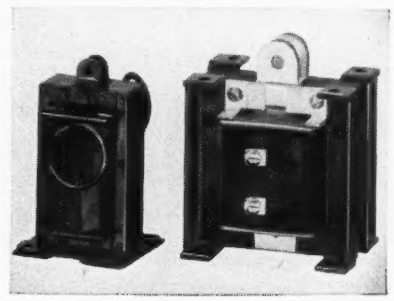
V-Belt with Cog Construction

HAVING a unique driving surface for either flat pulleys or V-grooved pulleys, a new type of V-belt is being offered by The Dayton Rubber Mfg. Co., Dayton, O. The belt is designed around a new type of cog construction principle on the inner surface of the belt, providing flexibility and longer belt life when flexing around small pulley diameters. For V-flat drives it applies the principle of increasing traction by means of a nonskid design. The cog construction provides greater grip and less

slip on smooth flat pulley surfaces and when riding in grooved pulleys, a greater gripping power with die-cut raw edge sides which present the same undistorted driving surface to the pulley regardless of side wear. The belt is available in all standard lengths in A, B, C, D and E cross sections, and in perfectly matched sets for multiple drive applications.

Solenoids for Wide Application

Wide range of sizes and strokes is included in a new line of solenoids made by Jefferson Electric Co., Bellwood, Ill. Small size and short stroke solenoids have numerous applications such as coin-operated or pin game devices; large size solenoids are used on machine tools, feed mechanisms, canning and bottling machines, textile machines,



damper control, electrically operated valves, ratchet mechanisms, brakes and clutch levers. By means of limit switches, mercury contacts, pressure switches, or pushbuttons, automatic operation can be effected. For continuous or intermittent duty, solenoids are available in voltages up to 550 volts, of 25, 50 or 60 cycles, and up to 550 volts direct current for intermittent duty only. Quiet operation is insured by proportioned shading coils and careful grinding of center pole and plunger faces.

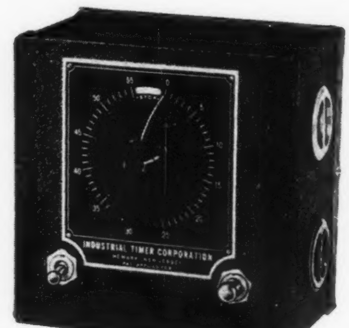
Drive with Planetary Gears

Introduced by the Process Equipment Division, H. K. Porter Co. Inc., 4949 Harrison street, Pittsburgh, a new agitator drive, because it utilizes planetary gearing has many advantages for the process industries. This gearing permits smooth operation, maximum reduction in a given space and minimum tooth load for given torsional effort. Although designed for standard NEMA frame motors, the new unit is adaptable to any type of motor. An unlimited ratio of reduction being permitted, any speed may be obtained in a complete size range from ½ to 50 horsepower.

Timer Buzzes Signal

Addition of a new buzzing timer to its line of synchronous motor-driven electrical instruments has been made by The Industrial Timer Corp., 101 Edison place, Newark, N. J. Timing pe-

riod is selected on a visible dial with pointer set to selected time calibration; the timer is switched on and instantaneous operation begins clocking of period. When time expires the BHD series shuts off timed operation automatically, and immediately resets itself for another timed operation of identical duration. The other types of timers start a signal which does not stop until timer is either



shut off or switched anew for a repeat operation. The BHR group of the BH series comes equipped with two sockets, one for a bulb to light at end of timed period, the other to accommodate a single wiring set-up to which single buzzer or unlimited numbers can be attached for remote warning. Among the applications for which the timers are suitable are baking ovens, metal-plating operations, vulcanizing, photographic developing, etc. Six standard models are available which extend time control over a wide range.

Porcelain Enamel Frits Offered

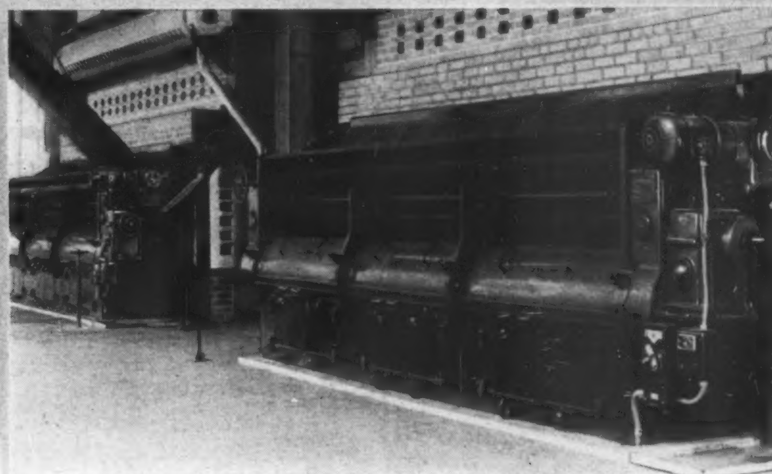
AMONG a new series of porcelain enamel frits is one known as Superpake, especially developed by the Porcelain Enamel & Mfg. Co., Baltimore, for a one-coat finish on refrigerators, washing machines and other volume production machines. The finish is an antimony-bearing frit and gives excellent coverage and high gloss. It is a white finish with low application weight and wide firing range.

Black Rustproof Finish

NEW rustproof finish—black on copper—has been announced by Alrose Chemical Co., Providence, R. I., as an addition to its Jetal process for blackening iron and steel. The new finish gives a black rustproof finish on copper by anodic oxidation. It can be applied on any metal that can be copper-plated. After a metal is preplated with copper, it is immersed in a special electrolytic oxidizing bath for two to five minutes, with the resulting velvety black surface of the Electro-Jetal. The surface is an excellent absorptive base for an after-treatment of oil, wax or lacquer which gives a nonporous, rustproof finish equivalent to ordinary nickle plate. The bath can be operated at any tem-

r Easier, Safer, Better Welding. The Allis Chalmers Welding Positioner handles rge fabricated steel shapes as heavy as 50 tons. It extends the range of jobs hich can be handled with automatic welding machines and makes hand welding asier and safer . . . Oilgear Transmissions make possible maintaining pre-deter-ined automatic welding speeds within close limits, they provide ample power to andle substantial overloads, and provide steplessly variable speeds to insure venness of automatic welds.

For a Wide Range of Speed Variation and Simple Control. Johnson and Jennings, Automatic Stoker Manufacturers, of Cleveland, Ohio, report that the Oilgear Units shown in the photograph are adjusted over a 16:1 ratio of speed variation and are doing a very successful job of stoker driving. Oilgear drives eliminate the necessity of special interlocking devices and are conveniently controlled from a central panel. Oilgear Transmissions are steadily gaining popularity with stoker manufacturers and users.



3 Outstanding Examples

OF THE MANY ADVANTAGES AWAITING YOU IN OILGEAR TRANSMISSIONS

For Far Greater Accuracy. *Below:* Ten Oilgear type DEC-1212 Transmissions are used to drive special Kant-Slip Continuous Form Printing Presses installed at the Standard Register Company of Dayton, Ohio. These presses must run with far greater accuracy than conventional web presses. The accurately controlled, smooth, cushioned power of Oilgear Transmissions protects register while adjusting and keeps stock spoilage down to a minimum. Presses can turn out 12,000 impressions per hour.

The unusual success of Oilgear Fluid Power Transmissions in so many *different* applications has led many machine designers to completely revise their opinions of what hydraulic transmission of power can accomplish. Oilgear Transmissions assure designers of smooth, uniform acceleration, high variable speed, accurate, sensitive control over a wide range, and efficient transmission of power in their machines; a combination which has resulted in safe, easy to operate automatic machines, higher speed, more productive machines and improved processes and products.

The list of special applications where Oilgear Transmissions have proved more productive and more economical is growing steadily. They are in use in practically every leading industry. If it's important that you get the most out of your production machines, it may well pay you to investigate the complete line of Oilgear Transmissions. Send now for case histories showing what Oilgear has accomplished either in your field or in a related field. Don't wait, but send now for this important information. Also send coupon for Bulletins 47000 and 60000 which give complete engineering data about the entire line of standard Oilgear Transmissions. THE OILGEAR COMPANY, 1305 W. Bruce Street, Milwaukee, Wisconsin.

Oilgear also manufactures Fluid Power Feeds, Pumps, Cylinders, Valves, Accessories, Horizontal and Vertical Broaching Machines, Horizontal and Vertical Presses, Custom Built Machines.

OILGEAR

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fluid power

TRANSMISSION

THE OILGEAR COMPANY, 1305 W. Bruce St., Milwaukee, Wis.
Please send me Bulletins 47000 and 60000 without obligation.

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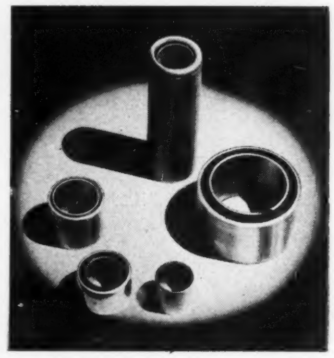
perature below the boiling point of water, using ordinary steel containers heated by steam coils. Work can be oxidized on racks, in bulk or in baskets.

Pre-Cast Bronze on Steel

STEEL backed, thin-walled leaded bronze alloy known as pre-cast bearing bronze on steel is announced by Johnson Bronze Co., 507 South Mill street, New Castle, Pa. Bushings and bearings made of pre-cast bearing bronze on steel combine bearing qualities of a bronze alloy with the strength of steel. Some advantages include a higher brinell hardness, a much greater resistance to pounding and shock, plus a comparatively low coefficient of friction. Bearings are available in a wide range of sizes and wall thicknesses, either plain leaded or graphited bronze. It is also possible to have the bearing ball-indented to retain grease. Any type of oil hole, groove or slot is easily incorporated. While the bronze on steel was primarily developed for bushings and bearings it also fills many other important industrial uses, such as applications requiring a flat bearing surface as plates, washers, etc. Bronze on steel in rolls up to 400 feet in length is also obtainable for stamping and forming. Maximum width of strip is 5½ inches.



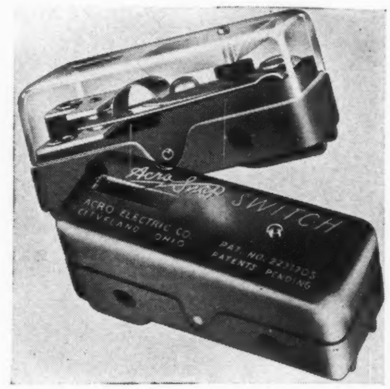
Vibration Insulators Offered

THREE additional new mountings of rubber and metal to isolate vibration have been announced by The B. F. Goodrich Co., Akron, O. Type 44, furnished with rubber either in shear or compression, is primarily for use with gas burners; Type 50 is specially engineered to isolate internal combustion engines against the transmission of torsional vibration; and Type 60 is for isolating vibration and noise of exhaust and intake fans. Type 44, in shear, has a maximum load of 10 pounds; deflection at that load being 5/32-inch and minimum disturbing frequency at that deflection, 1200 per minute. In compression the maximum load is 50 pounds, deflection at this figure ⅛-inch and the minimum disturbing frequency at 5/32-inch deflection is 1350 per minute. Type 50 compression construction will take care of minimum disturbing frequencies from 1200 per minute and up and is made in 35, 40, 50 and 60 durometer hardnesses. Type 60 is a compression mounting used where the lowest disturbing frequency is 1000 per

minute with a maximum recommended load of 240 pounds with 3/16-inch deflection at this loading.

Low Pressure Switch Announced

OPERATING on a new principle, a precision snap-action switch with a wide variety of modifications adapting it for wide range usage



has been announced by Acro Electric Co., 3179 Fulton road, Cleveland. Minimum of change in operation pressure to produce a direct and positive make and break is required. The switch which has an almost total absence of friction can be adapted to special shapes, sizes and positions, or built into almost any type of mechanism. It is single pole and is furnished with normally open, normally closed or double-throw contact arrangements. The standard models have the following alternating current ratings: 10 amperes, 115 volts; 5 amperes, 250 volts and 3 amperes, 450 volts. Pressure is ½ ounce to 14 ounces; weight, less than 1 ounce. Overall dimensions of the switch are 1 15/16 x 11/16 x 27/32 inches. To meet more unusual and exacting conditions, custom models are available. Simple construction of the switch permits extremely sensitive operating pressures.

Bearings for Rubber-Mounting

RUBBER-MOUNTED bearings with inner walls of a new, thinner material of either plain or graphited bronze on steel have been made available by Harris Products Co., 5408 Commonwealth avenue, Detroit. This type of bearing increases resistance to pounding and shock, and has a comparatively low coefficient of friction. It can be obtained in a wider range of sizes and shapes, as well as in sheet form to insure faster delivery. The required sizes are readily cut from the sheets, shaped and, in split form, can be quickly assembled with rubber insulating wall and outer wall. The latter may be any material required such as seamless steel, brass, stainless, etc. Increased shock resistance permits the material to be used in applications where excessively high impact and shock loads are present. With the thinner wall section the rubber wall surrounding it can be almost any

desirable thickness to handle the radial loads imposed or to compensate for misalignment, to absorb vibration and reduce noise.

Belt for High-Speed Drives

KNOWN as the Hevaloid belt, a new endless belt for high-speed precision work is being offered by the L. H. Gilmer Co., Tacony, Philadelphia. A unique process of impregnating the cotton pulling element with latex is said to form a material that is virtually homogeneous, resulting in special qualities of elasticity, strength and durability. The belt operates free from vibration inasmuch as it is made without lap, seam or splice and has a nonslip surface. This affords a high coefficient of friction. Practically a positive drive is thus produced, permitting lighter tension, decreased bearing load, making possible a higher and more uniform driven speed on drives such as in routing machines, winders, grinders and similar applications. Pliability for use over small pulley diameters and lightness in weight adapt it for operation at speeds up to approximately 9000 feet per minute.

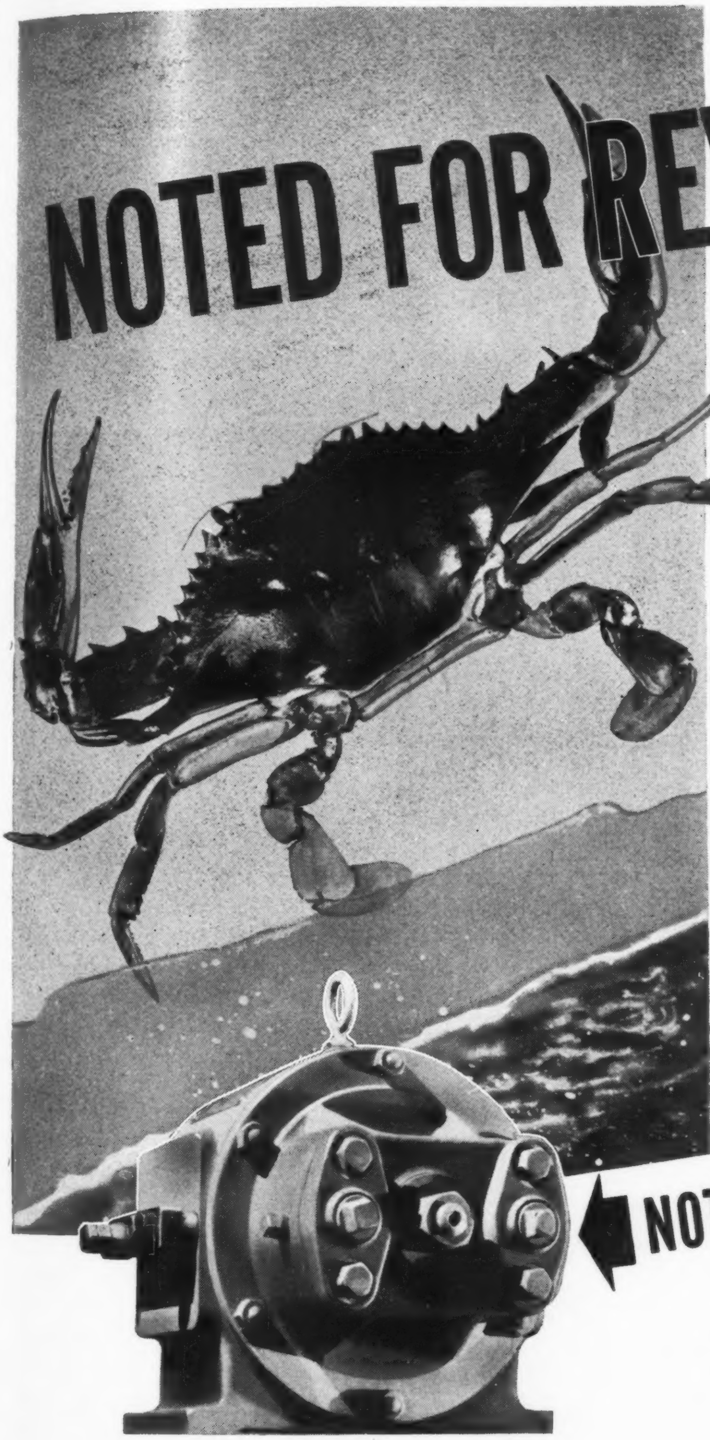
Sensitized Metal Developed

BECAUSE of the demand and the recent rulings on aluminum, the Republic Engineering Products Inc., 480 Lexington avenue, New York, has developed a new process whereby it is now able to supply other materials such as steel plate, brass, copper and various steel alloys, as well as plywood and plastics, coated with sensitized emulsion suitable for making photographic copies of all kinds and for all purposes. Some sensitized materials can be secured in thicknesses such as one-eighth or one-quarter inch cold rolled steel—the type used for making templates. In such cases the drawing is photographed directly on sensitized surface and then developed. The sensitized metal is suitable for nameplates, instrument dials, escutcheon plates, etc.

Lubricated Spiral Packing

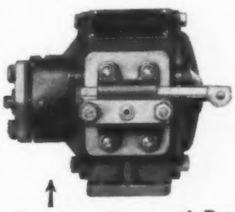
MADE of closely woven asbestos cloth, treated with a heat-resistant friction rubber, a new packing, furnished by Greene, Tweed & Co., 101 Park avenue, New York city, is adaptable for reciprocating rods and plungers of steam engines and pumps, air and gas compressors, and special machinery. This packing is known as the Palmetto high-pressure, spiral, asbestos packing. Center block is laid up against flat strip of heat resistant, resilient, red rubber, and cover or wear face is bonded to center to make a strong square corner fitting. It is spiraled on a mandrel and vulcanized so that it retains its shape. Graphite grease is forced into outer layer of packing by hot pressure operation, providing packing lubrication. The rugged construction of spiral packing with the lubricant decreases wear and increases the life.

NOTED FOR REVERSIBILITY

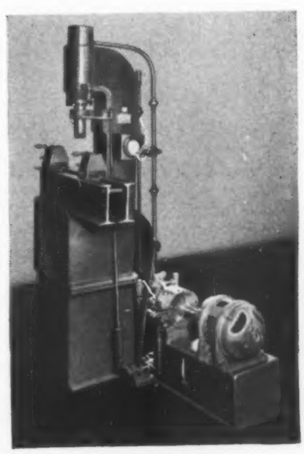


Hele-Shaw Fluid Power solves one of the tough jobs of machine design and operation—easy reversal. Fluid Power is oil under pressure from a Hele-Shaw Pump. It flows from the pump through pipes to the driven machine. The direction of this flow is changed by merely shifting the position of the guide rods on the pump. You can reverse the direction of flow easily and quickly. Fluid Power is smooth. It is shockless, cushioned by the oil medium. That's why, in presses, rams, or reciprocating devices, fluid power reversing is seldom equalled by any other method. And that's one of the reasons—among many others shown in our catalog — why Hele-Shaw Fluid Power is being used more and more.

NOTE IT FOR REVERSIBILITY



Hele-Shaw Pump with Type D Regulator—This, one of many Hele-Shaw regulators, reverses the direction of flow, gives any rate of discharge from zero to maximum in either direction.



20-TON RIFLE BARREL STRAIGHTENER, manufactured by the Watson-Stillman Company. Operated by a Hele-Shaw Pump equipped with a Type D Control.

THE
Hele-Shaw
Fluid Power Pump

OTHER A-E-CO PRODUCTS: LO-HED HOISTS, TAYLOR STOKERS, MARINE DECK AUXILIARIES

AMERICAN ENGINEERING COMPANY

2502 – ARAMINGO AVENUE, PHILADELPHIA, PA.

MEN *Of Machines*

THROUGH the engineering ranks to the top is, in brief, the career of J. C. Sharp, who recently was appointed vice president in charge of engineering of Hotpoint division of the Edison General Electric Appliance Co., Chicago. He joined the Hotpoint company in 1929 as assistant to the range engineer and in 1936 became range engineer. A year later he was made chief engineer, succeeding the vice president in charge of engineering. In his new appointment, Mr. Sharp will continue to carry out his duties also as chief engineer of the company.

Mr. Sharp received his training at Ohio State university, from where he received his engineering degree. He also spent a year at the United States Naval Academy.

AFTER twenty-two years with Westinghouse Electric & Mfg. Co., C. A. Powel has been appointed to the newly created position of manager of the headquarters engineering department, having supervision over four engineering departments. He formerly was manager of the industry engineering department of the company.

Mr. Powel was educated in Britain and Switzerland and had fourteen years of experience in electrical engineering in Europe and Asia before com-

ing to Westinghouse. He was graduated from the Institute of Technology of Canton Berne, Switzerland in 1905, and after six years of service in the British section of the internationally-known Swiss electrical firm of Brown Boveri & Co., was appointed resident engineer in Kobe, Japan. Returning to England in 1915 he joined the Civil Branch of the Ordnance department where he was assigned to inspection work at Woolwich arsenal. In 1916 he came to the United States on a war mission and later was assigned to the gage inspection department in New York city.

Honorably discharged from the British service in 1919, Mr. Powel joined the general engineering department of Westinghouse. In 1932 he was made manager of the central station engineering department and six years later was appointed head of the industry engineering department. He continued in this position until receiving his present promotion.

WELL known as a transformer engineer, Lee H. Hill has been appointed assistant manager of the Allis-Chalmers Mfg. Co. electrical department. Mr. Hill's experience includes two years as an instructor in electrical engineering at Cornell university previous to his becoming associated with Westinghouse Electric & Mfg. Co. as transformer design engineer. Later he was made engineer-in-charge of power transformer development for the Westinghouse company. For a number of years he was manager of the transformer division of the American Brown Boveri Co.—this before joining the Allis-Chalmers Mfg. Co. in 1931 when it absorbed Brown Boveri. Mr. Hill holds more than 40 patents relating to transformers and has written extensive and valuable contributions on the general subject. He has also been active in the affairs of the American Institute of Electrical Engineers.

♦

OTTO R. SCHOENROCK has been appointed chief engineer for the J. I. Case Co., Racine, Wis. Formerly vice president of the company, DAVID PRYCE DAVIES has been named consulting engineer for all the Case plants.

♦

RAYMOND C. BLAYLOCK has been named chief engineer at the new Port Columbus plant of Curtiss-

(Concluded on Page 110)

Morse High Speed Superdrives employ a recently developed and thoroughly tested principle to achieve their many advantages. With the same motor r.p.m.'s Morse Superdrives use larger sprockets with more teeth to get higher chain speeds—speeds up to a mile-a-minute and more!

At these high speeds, centrifugal force goes to work as Superdrives' helper, spreading the stresses so thin over chain and sprocket teeth that wear is reduced to a minimum. Power flows swiftly, smoothly, silently, and surely along the speeding chain, with power transmitting capacity actually rising faster than the speed of the chain.

Morse High Speed Superdrives offer great advantages—in the many applications to which it is suited—in long life, increased power transmitting capacity, strength, space economy, and dependability. Learn what they can bring to *your* high speed drives. Ask the Morse man near you, or write direct to Morse Chain Company, Ithaca.

SAME POWER FROM A DRIVE 1/3 THE WIDTH

Since power transmitting capacities are far greater in Morse Superdrives than in conventional drives, because of the greater chain speeds, Superdrives are narrower, more compact. For example: On a recently designed drive Morse specified 1″ pitch, 4″ wide chain to operate at 5100 feet per minute. Conventional design would have taken 1″ pitch, 12″ wide for the same capacity, the same motor speed, but with smaller sprockets and lower chain speed.

HIGH SPEED SUPERDRIVE

SILENT CHAINS ROLLER CHAINS FLEXIBLE COUPLINGS CLUTCHES

MORSE *positive* DRIVES

MORSE CHAIN COMPANY ITHACA N. Y. DIVISION BORG-WARNER CORP.

(*Concluded from Page* 104)

Wright Corp., working on the famous Curtiss Hawk and Curtiss dive bomber to be manufactured there. BRUCE G. EATON JR. has been appointed senior project engineer.

◆

JAMES DYKSTRA has been appointed assistant chief engineer of Olds Motor Works division of General Motors Corp. He will be in charge of chassis and body design, succeeding MAURICE A. THORNE. Mr. Thorne has been transferred to Detroit to be associated with C. L. MCCUEN, vice president of General Motors Corp., in charge of defense engineering.

◆

F. G. GARDNER of the engineering staff of Kellogg Switchboard & Supply Co., Chicago, has been selected to fill the position of acting chief engineer. He will be the head of the engineering department and research laboratory. GEORGE R. EATON, vice president in charge of engineering, is taking a leave of absence due to ill health.

◆

F. C. GLADDECK, for many years chief engineer with American Machine & Foundry Co., Brooklyn, has become associated with Machine & Tool Designing Co., designers and engineers, Philadelphia.

◆

H. H. ROTHROCK has assumed new duties as assistant to the chief industrial engineer, Carnegie-Illinois Steel Corp., Pittsburgh. Mr. Rothrock was former head of the industrial engineering department, University of Pittsburgh.

◆

RONALD KINNEAR of Niagara Insul-Bake Specialty Co., has been elected president of the Society of the Plastics Industry. E. C. MAYWALD, Chicago Molded Products Corp., is vice president; WILLIAM T. CRUSE, secretary-treasurer; and HENRY J. KASCH, Kurz-Kasch Inc., chairman of the board.

◆

C. S. BEATTIE has been appointed manager of engineering, Delta-Star Electric Co. W. O. HAMPTON, of the engineering department, is now chief design engineer, and S. C. KILLIAN, development and research engineer.

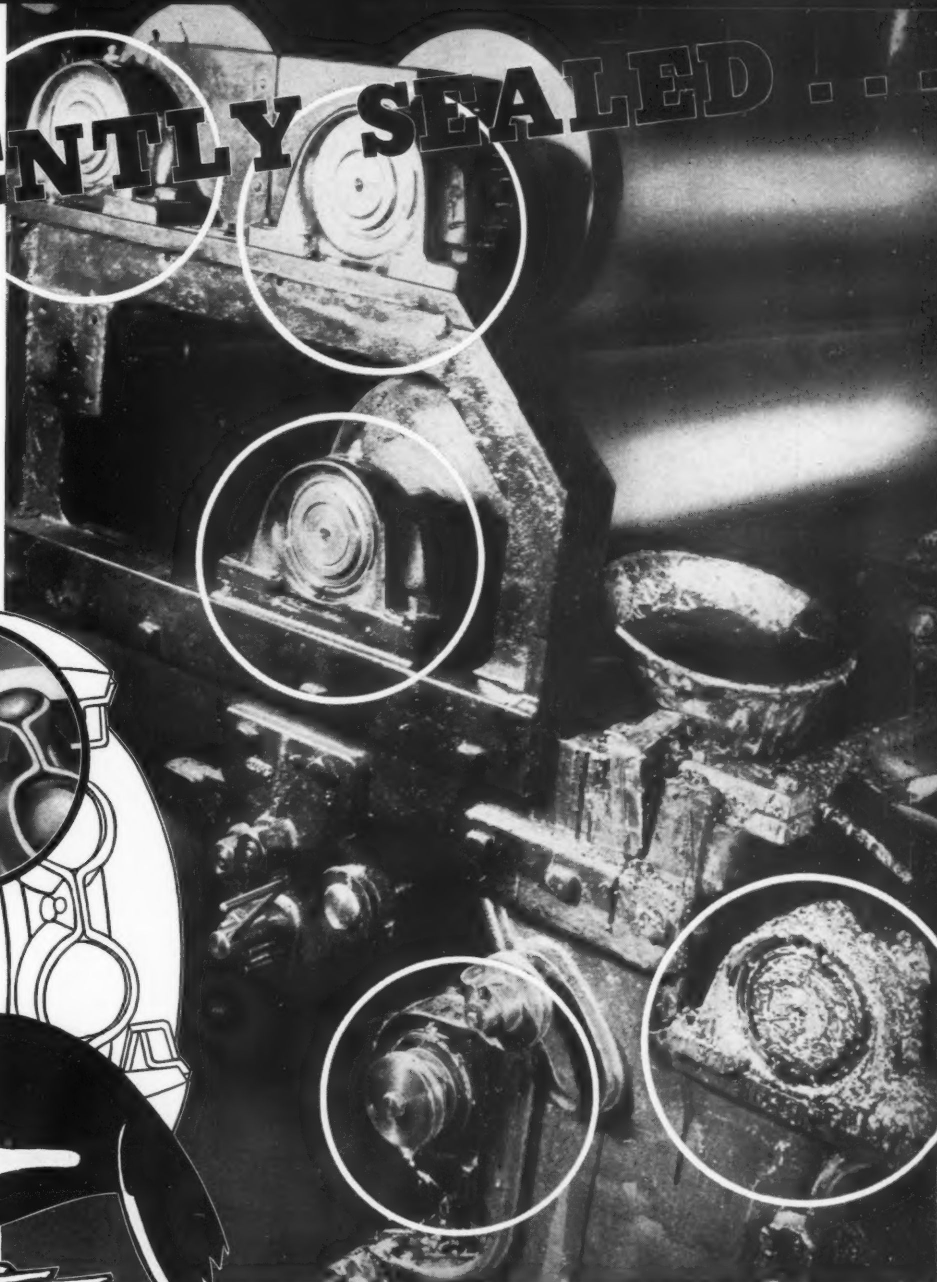
◆

DR. JOSEPH SLEPIAN has been elected to the National Academy of Sciences. He is associate director of Westinghouse Research Laboratories and is recognized as one of the world's outstanding electrical engineers. More than 200 patents stand to his credit.

◆

EDWARD T. MURPHY has been named president of the Air Conditioning and Refrigerating Machinery association. He is vice president in charge of marketing, Carrier Corp. Other officers elected are: C. E. WILSON, Worthington Pump & Machinery, first vice president; J. P. RAINBAULT, General Electric, second vice president; and P. A. MCKITTRICK, Parks-Cramer, treasurer.

PERMANENTLY SEALED . . .



to

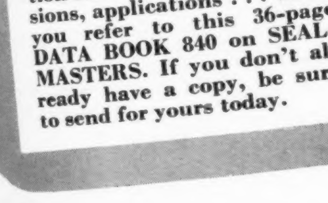
√ **Keep Out Dirt**

(even finest dust particles cannot enter this seal)

√ **Retain Lubricant Over Long Periods**

(Bearing Units also pre-lubricated prior to shipment)

FIND OUT HOW TO SAVE WITH SEALMASTERS . . .



You have complete information on construction, dimensions, applications . . . when you refer to this 36-page DATA BOOK 840 on SEALMASTERS. If you don't already have a copy, be sure to send for yours today.

Dirt, carbon and fine particles of dust all over this machine do not affect the efficiency of these SEALMASTER Bearing Units . . .

. . . because the advanced design of *patented* seal in SEALMASTERS does effectively the two-fold job of *keeping dirt out* of bearing while *retaining lubricant over long periods* . . . minimizing maintenance attention.

In the SEALMASTER design, you secure *self-alignment without possible distortion of the seal* . . . for seal is contained within bearing proper, independent of housing.

Steel flingers, attached to inner race ring, have special sealing felts impounded on inner faces. These felts, rotating in groove of inner seal, are assembled with close running clearances but without interference, and are *free-running*.

This seal design results in *long bearing life* and *care-free service*.

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STEPHENS-ADAMSON MFG. CO.

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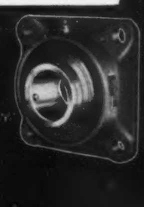
SEALMASTER BALL BEARING UNITS

Pillow Blocks, Flange Units



Take-Ups, Take-Ups in Frames

Cartridge Units, Hanger Bearings

Extended Inner Ring Bearings

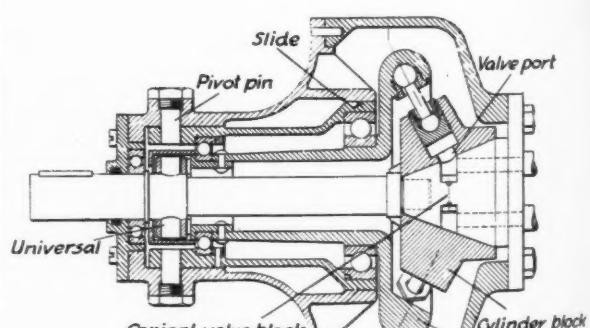
Noteworthy PATENTS

Leakage Unaffected by Pressure

LEAKAGE in hydraulic radial piston pumps and motors is ordinarily a direct function of the fluid pressure. This is a result of clearances necessarily existing between relatively moving fluid-handling members. Such clearances act as orifices, permitting fluid flow at a rate proportional to upstream pressure assuming downstream pressure is constant and low.

In a patent, assigned to Manly Corp., a pump is disclosed in which increases in pressure serve to reduce operating clearances between the cylinder and valve blocks so that leakage rate can be designed to be maintained constant over the operating pressure range. This is accomplished by positioning the cylinders so that fluid pressure has a component in the direction of reducing clearance. In addition, provision is made for adjusting the volumetric displacement through zero to a maximum in either direction.

Mounted rigidly in housing is a frustro-conical valve block provided with two diametrically opposed chambers, one for inlet, the other for discharge. Registering with these chambers are the



Inclination of cylinder axes provides a component of pressure in the direction of reducing clearance in hydraulic pump or motor

valve ports of each cylinder formed in the cylinder block. The entire cylinder block assembly is driven through dog teeth formed on the shaft shoulder which abuts against the left face of the block.

One ball end of each connecting rod is attached to a piston; the other ball end to the crank disk. The reduced diameter of this disk extends to the

left and is supported in two ball bearings, the inner races of which rotate with the disk. Outer races are housed in a rectangular slide member.

The left extension of the small end of the disk ends in a fork, forming the driven member of a universal. The driving member consists of a pin driven through the drive shaft. The rectangular slide member is hinged at its left end on two pivot pins.

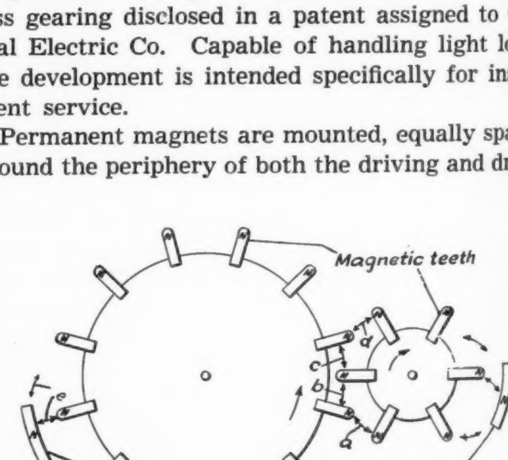
Fluid pressure existing between the valve and cylinder block as well as that exerted on the cross-sectional area of the valve ports, tends to thrust the cylinder block to the left, increasing clearances and hence leakage. However, since the port diameters are smaller than the cylinder diameters and because the axes of the cylinders are inclined to the axis of rotation, the leftward thrust on the cylinder block is opposed by a thrust to the right which is a function of the difference in port and cylinder areas and the sine of the angle of inclination from the straight radial. Thus by properly selecting the angle of inclination of the cylinder axes as well as the diameters of cylinders and ports, the pressure of the cylinder block on the valve cone can be made to increase with the pressure, thereby reducing clearance and leakage.

Adjustment of the eccentricity of the disk with respect to the cylinder block is achieved by pivoting the disk and bearing assembly about its pivot pins.

Drives Without Contact

INGENIOUS utilization of the repulsion effect of permanent magnets characterizes the frictionless gearing disclosed in a patent assigned to General Electric Co. Capable of handling light loads, the development is intended specifically for instrument service.

Permanent magnets are mounted, equally spaced, around the periphery of both the driving and driven



Repulsion effect of permanent magnets results in the practical elimination of all friction and other mechanical losses

non-magnetic disks. This mounting is accomplished in such a manner that, should the driven gear be for any reason overloaded, the magnets of the two members will come into actual contact with each other. Thus the relationship, insofar as tooth engagement is concerned, of one gear to the other is positively maintained.

Considering the left-hand member in the illus-

tration as the driver it is apparent that, as the load on the driven gear increases the space labelled b will decrease; that labelled c will increase; and d and a will change only slightly. Hence repulsion force tending to produce rotation of the driven member increases while similar forces tending to restrict rotation decrease or remain substantially unchanged.

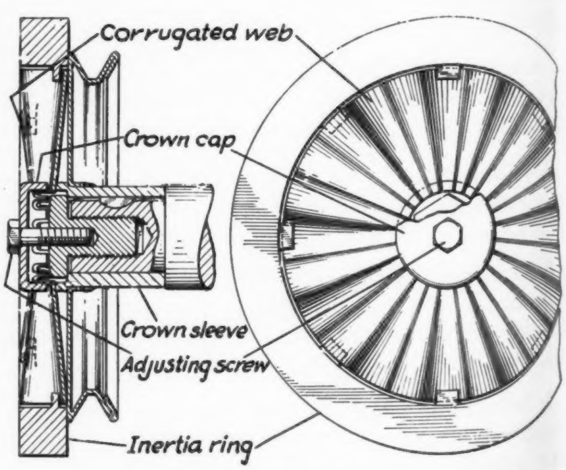
In order to minimize the radial thrust on the two gears which results in an increase in load on the bearings, compensator magnets are installed. These may be adjusted to balance accurately the gear separating forces. Compensators are shown installed somewhat below the line of centers of the two gears. The purpose of such an installation is to compensate for, in addition to side thrust, the weight of the gears on their bearings. Thus the device may be set up so that bearing loads are essentially nonexistent.

Web Suppresses Vibration

HEAVY rotating masses such as flywheels, when rigidly affixed to shafts subject to torsional vibration as in internal combustion engine crankshafts, set up high stresses in these shafts. In a patent assigned to General Motors Corp., rigid spokes are replaced by a flexible web which prevents transmission of vibration to the flywheel.

Web is of sheet metal corrugated radially. It is held at its outer edge to the flywheel by means of alternately staggered lugs on the inside of the wheel. These lugs engage the corrugations and effect a driving relationship between the web and wheel.

The inner or hub section of the web is clamped between two crown-shaped members. One of these



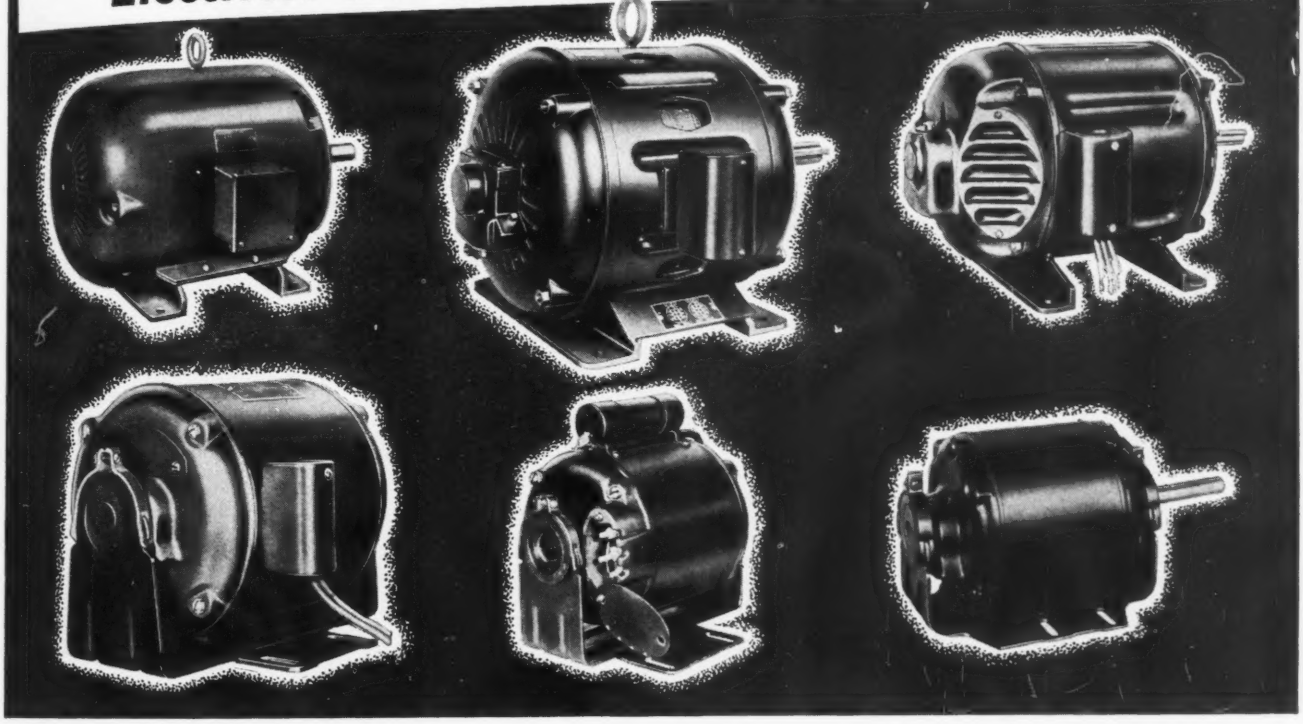
Radically corrugated web prevents transmission of torsional shaft vibrations to the flywheel

is a sleeve clamped against the shaft shoulder; the other is a cup held in place by an adjusting screw.

By tightening the adjusting screw increased stress is imposed on the web, thus increasing its natural vibrating frequency. Also, a pulley

WAGNER MOTORS ARE BUILT TO MEET EVERY REQUIREMENT

... a Wide Range of Types and Sizes with varied Electrical and Mechanical Characteristics ....



A good motor is not enough—it must be the right motor for the job. In order to properly select a motor, the following points should be considered.

Load Cycle ... What maximum and minimum horsepower is involved, and what is the probable duration of each?

What are the maximum starting torque requirements?

Is the duty cycle continuous or intermittent, and what method of control and overload protection is contemplated?

Power Supply ... A. C. or D. C., and frequency if A. C. ... Voltage ... Phase ... Special starting current limitations, if any, imposed by the power supplier.

Speed Characteristics ... Single constant speed ... Variable or multispeed.

Mechanical Construction ... Is open type motor acceptable ... or should Splash proof, Totally-enclosed fan-cooled, or Explosion-proof motor be used?

A Wagner, factory-trained, sales-engineer is ready to assist you in selecting the proper motor for your specific requirements.

WRITE FOR BULLETINS MU-177, MU-179 and MU-182

Diversified Types: It's easy to select a WAGNER motor that exactly fits the job because there is such a wide range of types and sizes from which to choose, each one designed and built for a particular kind of service.

Modern Design: Wagner motors are neat and compact, up-to-the-minute in design, and harmonize with the most modern machinery and equipment.

Dependable Performance: Wagner motors are well engineered and sturdily constructed, making them absolutely dependable even when operating under the most severe service conditions. A fifty-year reputation for satisfactory performance is your assurance that the best buy in motors is Wagner.

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M41-18

Wagner Electric Corporation

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MOTORS • TRANSFORMERS • FANS • BRAKES

mounted on the shaft bears against the face of the wheel, preventing it from wobbling and imposing a friction drag which reduces forced vibrations which may be transmitted through the web.

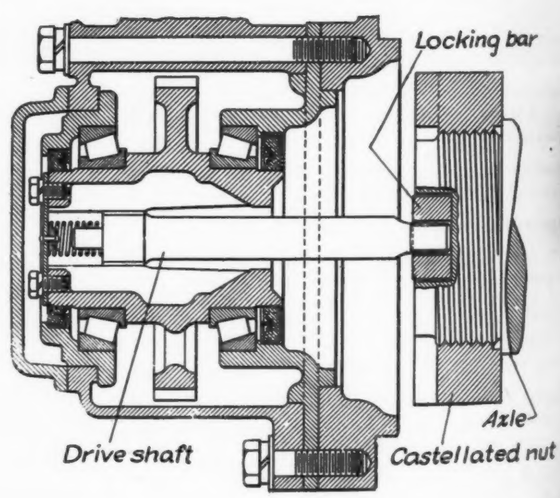
In assembling the device the center of the web is displaced with respect to its outer periphery so that it assumes a conical shape. In this shape it may be inserted inside the ring and, when released, will properly engage the lugs.

Drive Has Full-Floating Shaft

ASSEMBLY and disassembly of speed-responsive drive mechanism is facilitated by a design disclosed in a patent assigned to The Union Switch and Signal Co. Designed to operate a governor or other equipment which may be synchronized with an axle of a railway train, it may be easily and quickly connected or disconnected without detaching the entire assembly from the axle journal bearing housing.

Screwed over the threaded end of the car axle is a castellated nut between the lugs of which is constrained a locking bar. Two cap screws tapped into the end of the axle serve to hold the bar in place. In the midpoint of the bar is a rectangular hole designed to accommodate the substantially rectangular end of the drive shaft.

The other end of the shaft is square in form and fits in a similar square hole in the gear hub. Since the direction of drive is usually constant for long periods the fits of the ends of the shafts



Axle-driven device for attachment to railway car axles permits the driving of governors or other speed responsive equipment on each car

can be quite free to accommodate nominal misalignments of the drive with respect to the axle. Similarly, spring loading the end of the shaft permits free axial float of the axle within the limits imposed by its bearings.

By simply removing the cap screws holding the end plate on the left side of the housing, the spring and shaft may be removed. Thus the drive may be disconnected from the axle without disturbing the gear or bearing assembly.


"WE WANTED A COMPLETE, ALL-PURPOSE PROCESS OF MAKING PRINTS...THAT'S WHY WE GOT AN OZALID MACHINE"

Ingenious Mechanism in Jig Grinder Design

(*Continued from Page* 48)

is obtained by driving this shaft through a pair of adjustable pitch V-belt pulleys from the motor-driven driveshaft shown in the upper section of *Fig.* 5.

As previously stated fine adjustments of the position of the grinding wheel assembly on its cross slide must be made while the machine is running. For larger, rough adjustments the machine must be stopped.

This is made possible by means of a plunger concentric with the vertical slide, *Figs.* 2 and 5, which has, formed on its lower end, a tapered cam surface. One end of the lever hinged in the slide block coacts with this cam surface. When the plunger is moved down, the end of the lever is forced to the right. Since the wheel assembly is attached to a slide pinned to the opposite end of the lever, the grinding wheel is thus moved to the left.

By tapping radially the pin by which the wheel assembly is held to the slide and screwing into this hole the capscrew shown, large rough adjustments of the slide can be made while the spindle is stationary.

The plunger extends up to the top of the vertical slide where it terminates at a linkpin similar to the universal drive link. The plunger link is shown inside the universal link in *Fig.* 5. Fastened to the other end of the link is a rod which passes through and is concentric with the control spindle.

In order that adjustments may be made while the spindle is rotating, a pin is driven radially through the upper end of the plunger control rod, passing through an axial milled slot in the control spindle. The ends of this rod bear against a thrust bearing carried in the micrometer adjustment nut. By turning this nut the adjustment plunger is moved up or down, shifting the grinding wheel assembly either to the right or left.

V-Belts Drive Vertical Feed

Vertical feed of the grinding head is a function of the speed of rotation of the spindle. The vertical shaft which carries the sprocket for driving the spindle carries also a variable speed V-belt pulley to drive, through a second pulley, the vertical feed shaft shown in *Fig.* 5. At the lower end of the vertical shaft a spur pinion is mounted in engagement with two diametrically opposed gears shown in *Fig.* 6. Each gear drives a worm shaft, one in one direction and the other in the opposite.

Wormwheels shown in the upper view of *Fig.* 7 are driven in opposite directions by the worms. The shaft to which the wheels are bushed drives, through intermediate gearing, pinions meshing with racks formed on guide bars which raise and lower the vertical slide, providing the feed.

Driving engagement between the wormwheels

and the shaft in the upper view of *Fig.* 7 is obtained by a dog clutch mounted between the wheels and splined to the shaft. The trigger mechanism by which the shifting of the clutch from engagement with one wormwheel to engagement with the other is shown in the lower view of *Fig.* 7.

As illustrated, the clutch-actuating slide is in midposition corresponding to the position of the clutch itself in the upper view. Normally, however, the slide is in either the extreme right or left position so that the catches on the slide ends are engaged by either the right or left latches.

Feed Reversed by Trigger

To facilitate the understanding of the design of this mechanism let it be assumed that the slide is moved to the extreme right position, engaging the right-hand latch. The pin through the midpoint of the slide which engages the fork in the stem of the inverted T-shaped rocker thus rotates the rocker clockwise about its hinge pin. This tends to compress the right-hand compression spring.

The housing containing the two actuating springs as well as the trip adjusting setscrews is also hinged on the same pin as the T-shaped rocker. An arm extending from the left of this housing serves not only as a handle for manual operation but also as a lever operated by one of two adjustable limit blocks attached to a rod which moves vertically with the vertical motion of the grinding head.

With the clutch-actuating slide in its right-hand position, the grinding head moves down. As it approaches the bottom of the hole being ground, the upper limit block engages the operating arm of the trigger mechanism, rotating the spring housing counterclockwise about its hinge pin. This rotation serves to compress further the right-hand actuating spring, exerting, through the rocker, a considerable thrust on the actuating slide tending to move it to the left. As the spring housing is rotated further counterclockwise the right-hand trip-adjusting setscrew rotates the right-hand latch clockwise about its pin until the latch releases the slide.

Spring Shifts Clutch

Energy stored in the right-hand spring then instantly drives the slide to the left, shifting the clutch from one wormwheel to the other and thereby reversing the direction of vertical traverse of the grinding head. Also, the left-hand latch is engaged. The mechanism is thus cocked to reverse again the direction of traverse of the grinding head when it reaches its upper limit.

All too often the ultra-precision work of the toolmaker resembles in its consumption of time and amount of tedious hand labor the work of the guild craftsmen of the renaissance. In contrast, the jig grinder discussed herein embodies sufficient automatic controls of all operating functions so that, once set up, the machine runs itself and produces taper, finish and dimension to a degree of accuracy heretofore available only at excessively high cost.

Theory of Elasticity in Practical Design

(*Continued from Page* 55)

broken down into two parts, as in the following:

$$\int_{-a}^{a} q \log (z-x)^2 dz = \int_{-a}^{0} q \log (z-x)^2 dz + \int_{0}^{a} q \log (z-x)^2 dz$$

The effect of the integration from $-a$ to 0 on the point R, located $z = +x$ from the origin, is the same as the integration from 0 to $+a$ on a point R', located at $z = -x$ from the origin, as shown in *Fig.* 85. This change in limits replaces $(z-x)$ with $(z+x)$, thus

$$\int_{-a}^{0} q \log (z-x)^2 dz = \int_{0}^{a} q \log (z+x)^2 dz$$

Putting this another way, the strain at R due to the portion of the load from $-a$ to 0, is the same as the strain at R' due to the portion from 0 to $+a$.

This then gives for the second integral in Equation 141

$$\int_{-a}^{a} q \log (z-x)^2 dz = \int_{0}^{a} q \log (z^2-x^2)^2 dz$$

Substituting this and Equation 143 back into 141 gives

$$u_y = -\frac{1-\lambda^2}{\pi E}\left[\frac{1}{2}\pi a q_0 \log C^2 - \int_{0}^{a} q \log (z^2-x^2)^2 dz\right] \qquad (144)$$

Differentiating Equation 144 with respect to x and substituting q from 142

$$\frac{du_y}{dx} = \frac{4(1-\lambda^2)q_0 x}{\pi E a}\int_{0}^{a}\frac{\sqrt{a^2-z^2}\,dz}{x^2-z^2}$$

The integrand in the above may be rationalized by introducing a new variable t, such that

$$t = \frac{z}{\sqrt{a^2-z^2}}$$

which gives

$$\frac{du_y}{dx} = \frac{4(1-\lambda^2)aq_0 x}{\pi E(a^2-x^2)}\int_{0}^{\infty}\frac{dt}{(1+t^2)(m^2-t^2)}$$

where $m = x/(a^2-x^2)^{1/2}$. Separating the factors

in the denominator of this integrand gives

$$\frac{du_y}{dx} = \frac{4(1-\lambda^2)q_0 x}{\pi Ea}\left(\int_0^\infty \frac{dt}{1+t^2} + \int_0^\infty \frac{dt}{m^2-t^2}\right) \quad \ldots\ldots (145)$$

The first integral in Equation 145 gives

$$\int_0^\infty \frac{dt}{1+t^2} = \frac{1}{2}\pi \quad \ldots\ldots (146)$$

The integrand in the second integral has a discontinuous point at the value $t = m$, as shown by its graph in *Fig.* 86. To evaluate it is necessary to integrate separately over the regions 0 to $m - \epsilon$, and $m + \epsilon$ to ∞, and then allow ϵ to diminish to zero. Thus

$$\int_0^\infty \frac{dt}{m^2-t^2} = \lim_{\epsilon \to 0}\int_0^{m-\epsilon} \frac{dt}{m^2-t^2} - \lim_{\epsilon \to 0}\int_{m+\epsilon}^\infty \frac{dt}{t^2-m^2}$$
$$= \lim_{\epsilon \to 0}\frac{1}{2m}\left[\left(\log\frac{m+t}{m-t}\right)_0^{m-\epsilon} + \left(\log\frac{t+m}{t-m}\right)_{m+\epsilon}^\infty\right]$$

Lower limit of the first parenthetical expression is log 1, which is zero. The limit as $t \to \infty$ of $(t + m)/(t - m)$ is also 1, giving zero for the upper limit in the second. Therefore

$$\int_0^\infty \frac{dt}{m^2-t^2} = \lim_{\epsilon \to 0}\frac{1}{2m}\left[\log\frac{2m-\epsilon}{\epsilon} - \log\frac{2m+\epsilon}{\epsilon}\right]$$
$$= \lim_{\epsilon \to 0}\left[\frac{1}{2m}\log\frac{2m-\epsilon}{2m+\epsilon}\right]$$

The limit of the quantity in the brackets is here also log 1, so that

$$\int_0^\infty \frac{dt}{m^2-t^2} = 0$$

Substituting this and Equation 146 back into 145

$$\frac{du_y}{dx} = \frac{2(1-\lambda^2)q_0 x}{Ea} \quad \ldots\ldots (147)$$

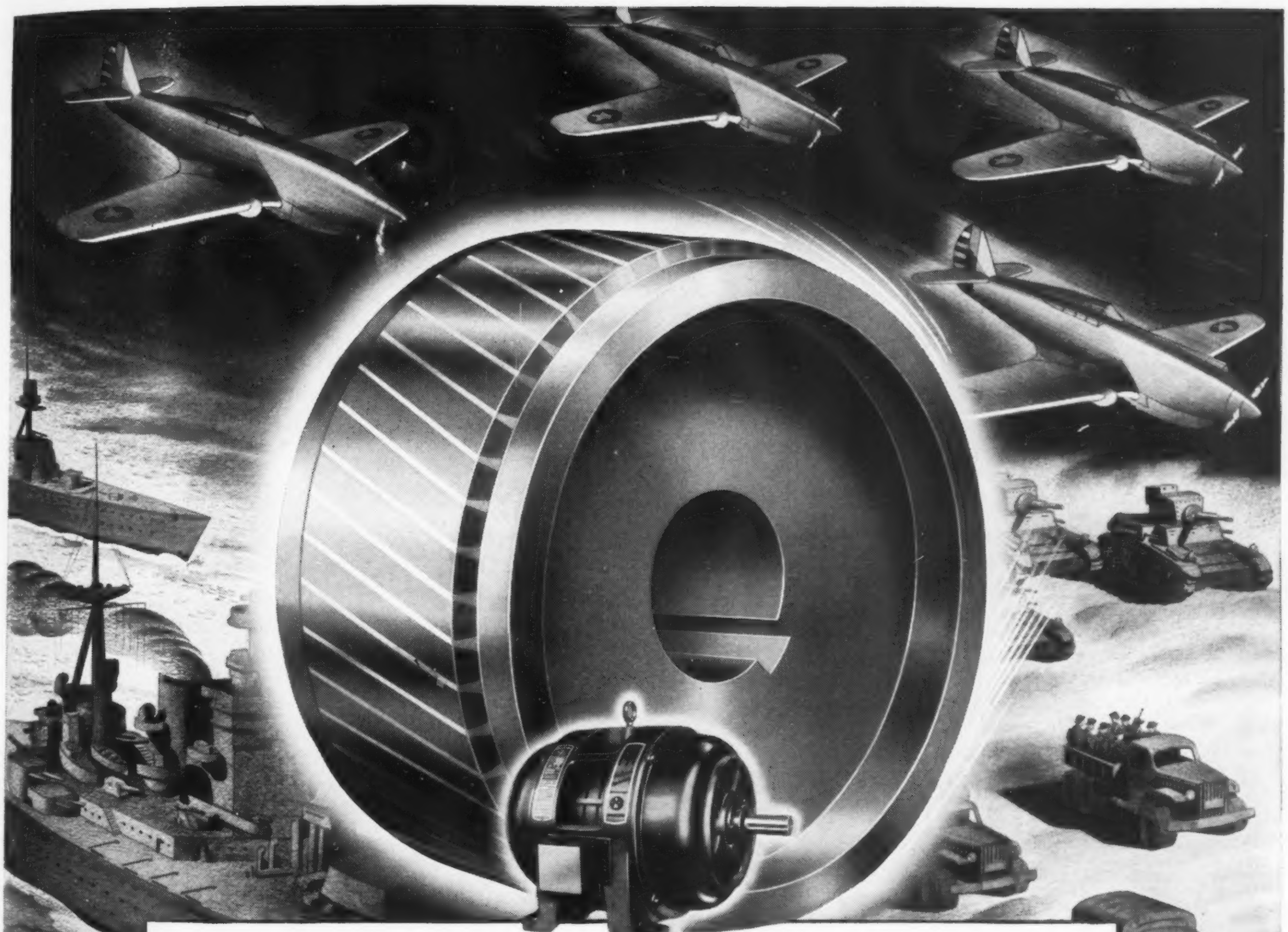
Integrating Equation 147 gives for u_y

$$u_y = \frac{(1-\lambda^2)q_0 x^2}{Ea} + C_4$$

Letting the strain at the origin be $-e$, that is $u_y = -e$ when $x = 0$, gives $C_4 = -e$. Substituting in the above gives finally for u_y

$$u_y = -\left[e - \frac{(1-\lambda^2)q_0 x^2}{Ea}\right] \quad \ldots\ldots (148)$$

Now u_1 and u_2 in Equation 137, the strains on the



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Copperspun
Patented

FAIRBANKS-MORSE MOTORS

DIESEL ENGINES ELECTRICAL MACHINERY RAILROAD EQUIPMENT WASHERS-IRONERS STOKERS
PUMPS FAIRBANKS SCALES WATER SYSTEMS FARM EQUIPMENT AIR CONDITIONERS

roller surfaces, may be obtained from 148 by changing the sign to cover the change in the direction of the Y axis, and adding the proper subscripts. Letting

$$N=\frac{1-\lambda_1^2}{E_1}+\frac{1-\lambda_2^2}{E_2}\qquad\qquad(149$$

we obtain from Equation 137

$$(e_1+e_2)-\frac{Nq_0}{a}x^2=e-\beta x^2$$

which shows the form of Equation 137 satisfied, establishing the elliptical distribution given by 142. Also, by equating the coefficients of x^2 and solving for a

$$a=\frac{N}{\beta}q_0$$

This completes the solution for the contact load distribution. The next article will determine the subsurface stress in terms of q_0. The evaluation of q_0 in terms of the total load will then complete the solution. This will be followed with a number of practical examples.

Selecting Special Motors

(*Continued from Page* 73)

design is supplied in types that are adjustable while running or set permanently to a predetermined speed. Adjustment on the first is made by stopping the motor and adjusting a small setscrew which serves as a stop for the inner contact, while the latter is provided with a mechanical linkage which leans against a stop movable in an axial direction to adjust over wide ranges of speed.

Under best conditions of operation, with constant load, speeds can be held to variations of less than plus or minus 1 per cent with a plus or minus 10 per cent variation in line voltage. *Figs.* 6 and 7 show the circuit and general performance of governor motors designed to operate at speed ranges of 2000 to 3000 revolutions per minute.

Gives Better Performance

The resistor and capacitors which are connected in parallel with the contacts are used to obtain a clean break at the contacts, giving smoother motor operation and longer contact life. The capacitor, which is placed in parallel with the governor contacts, absorbs the inductive energy generated when the circuit is broken and subsequently discharges it through the resistance rather than have it dissipated in arcing at the contacts. The resistor can be adjusted to take a considerable portion of the load, thus reducing the number of makes and breaks and increasing the contact life. Motor starts and accelerates as a conventional series motor. Upon reaching a predetermined speed the contacts open, cutting off motor current so that the speed drops and the contacts close again and per-

mit a repetition of the acceleration period. This cycle is repeated at high frequency, holding the motor at an average speed between the maximum and minimum. The design of the governor weight system, the weight of the movable member, rotor inertia, the connected load and ratio of available to load torque all effect the speed limits at which the motor operates and the variation to be expected from load or line changes.

The governor permits the motor to take power from the line more rapidly on heavy loads than at light loads and the average input is therefore controlled by the load imposed upon the motor. The motor can be designed with outputs in excess of those required by the normal operating conditions to provide a reserve available for momentary peak conditions and fast acceleration.

Has Shortened Life

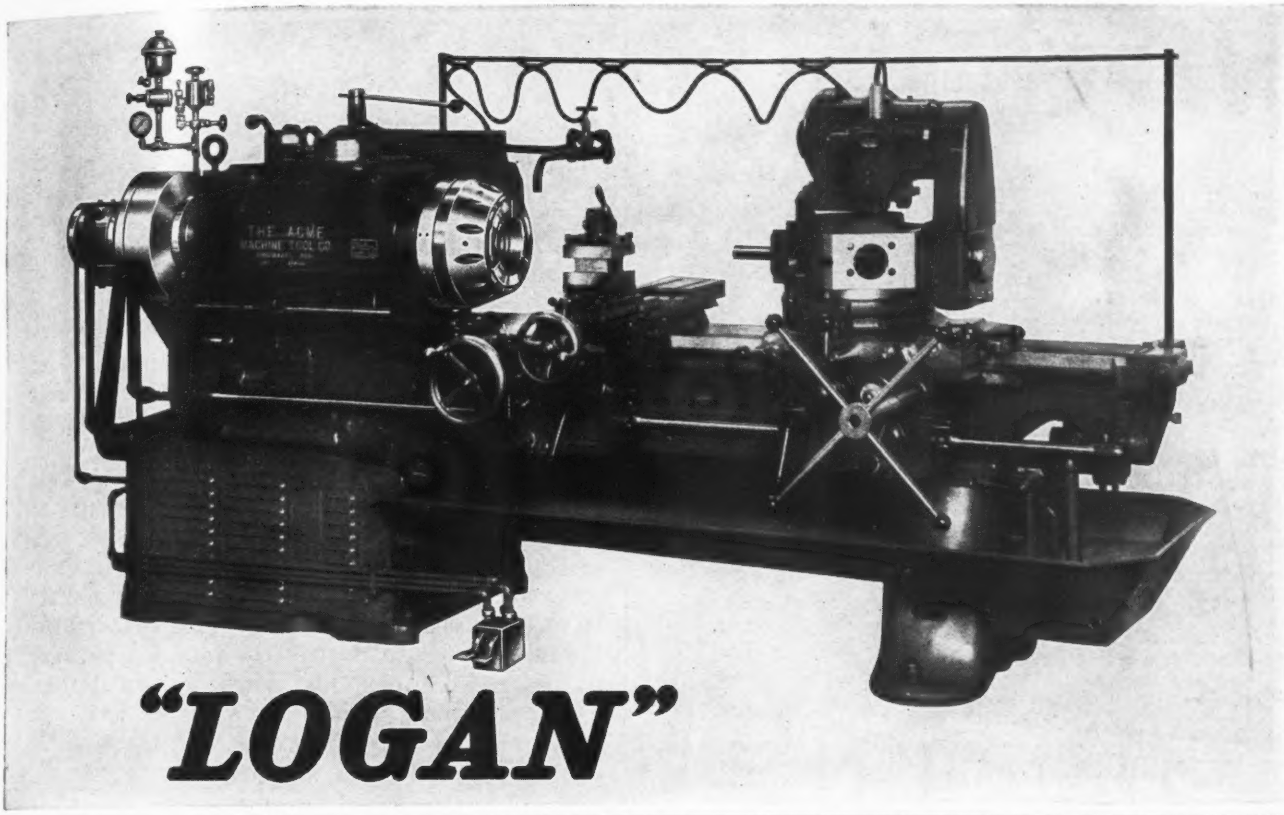
Addition of a governor to the series motor changes remarkably the performance characteristics but retains the desirable features of torque and acceleration. While brush life is reduced somewhat by the addition of a governor, the slower speeds at which these motors operate tend to overcome this loss, giving expected values of from 2000 to 3000 hours.

Governors are operative on alternating current and direct current but special arrangements are required for direct current operation to avoid transfer of metal on the contacts. This is done by splitring collectors which reverse the current through the contacts and parallels normal alternating current operation.

It should not be overlooked that the life of series motors is shorter than that of induction motors, efficiency is lower and radio interference is produced. Were it not for the universal characteristics, high torque and light weight of the series motor, the induction type may long ago have made a bid to replace it. In fact, even though series motors, refined by the arrangements discussed in the foregoing, have given a good account of themselves, certain of the office device and projector applications already are turning to the use of small induction motors.

In conclusion, it is hoped that this article sufficiently indicates the importance of close co-operation between the machine designer and motor manufacturer to the end that the best devices may be produced to keep in step with constantly changing designs and requirements.

TO PROVIDE useful information on physical, chemical and metallurgical properties of welded carbon and alloy steel tubing in a convenient, compact form, the Formed Steel Institute has published a "Handbook of Welded Steel Tubing." Engineering and fabricating data that are thought to be most essential to correct use are summarized for the design engineer.


"LOGAN"

Viewpoints

(*Concluded from Page* 79)

diagram with momentary torques as ordinates and rotation in radians as abscissas. The above integration can, of course, be performed analytically or graphically, or by simple tabular arithmetic.

To find the elapsed time of the acceleration the calculus would require an easily integrable expression for the relation between the momentary torque and the traversed angle. Such an expression does not exist for many kinds of machines, such as alternating-current and direct-current motors and generators. However, tabular arithmetical summation can be used as follows:

From the angular work diagram, the total stored kinetic energy at each successive radian of angular traverse is quickly found, being the area of the diagram up to that point. Hence the angular velocity at each radian is found, and hence the elapsed time to traverse each successive radian. Simple addition then gives the total elapsed time of the acceleration. This tabular method is generally applicable to any machine for which the angular work diagram can be obtained, either analytically or from a strip chart.

It is felt that the above comments should make more useful the many helpful suggestions contained in the article cited, particularly those taken from the practical experience of the authors in the application of motors to severe services.

—R. E. BRUCKNER
Manager of Machine Design
Kimble Glass Co.

". . . let's do it right"

To the Editor:

I am one of the many designers who would like to see the elimination of the sloppy methods sometimes existant in the design of machines. It is indeed appalling to see what goes on in many plants. Scores and scores of designs have 50 to 200 per cent more material than required. Once in a while it appears that an executive engineer of a company tries to overcome breakdowns by adding unnecessary weight to all possible parts, thus creating incompetent designs through lack of technical knowledge.

Wasting material is a crime and no management should allow any such condition. To save materials, we should all apply correct engineering methods in design.

Let's do it right for a change!

—R. W. C.
Cleveland

PLASTICS, compounded from oil refinery gases formerly used as furnace fuel, are easily molded and machined and are comparatively light in weight. These by-product materials are low in cost and will be extremely useful to conserve strategic materials needed for defense.

Meetings and Expositions

Sept. 8-12—

American Chemical Society. Semiannual meeting to be held at Chalfonte Haddon Hall, Atlantic City, N. J. Dr. Charles I. Parsons, Mills building, Washington, is secretary.

Sept. 25-26—

Society of Automotive Engineers. National Tractor meeting to be held at Schroeder hotel, Milwaukee. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary.

Sept. 23-26—

Association of Iron and Steel Engineers. Annual meeting and exhibit to be held at the Statler Hotel, Cleveland. Brent Wiley, 1010 Empire building, Pittsburgh, is managing director.

Sept. 28-Oct. 2—

American Mining Congress. Annual metal mining convention and exposition to be held at Fairmont Hotel, San Francisco, Julian D. Conover, 309 Munsey building, Washington, is secretary.

Oct. 6-10—

National Restaurant Association. Annual meeting and exhibition to be held at the National Restaurant Mart, Chicago. Frank J. Wiffler, 666 Lake Shore drive, Chicago, is secretary.

Oct. 6-10—

National Safety Council. Annual meeting and exposition to be held at the Stevens Hotel, Chicago. W. H. Cameron, 20 North Wacker Drive, Chicago, is director.

Oct. 6-11

Exposition of Power and Mechanical Engineering to be held at the International Amphitheatre, Chicago. Charles F. Roth, Grand Central Palace, New York, is manager.

Oct. 13-14—

Steel Founders' Society of America. Fall meeting to be held at The Homestead, Hot Springs, Va. Raymond L. Collier, 920 Midland building, Cleveland, is secretary.

Oct. 13-16—

Marking Device Association. Annual meeting to be held at the Palmer House, Chicago. Homer G. Klene, 431 South Dearborn street, Chicago, is secretary.

Oct. 16-18—

American Society of Tool Engineers. Semiannual convention to be held at Royal York hotel, Toronto, Canada. Ford R. Lamb, 2567 West Grand boulevard, Detroit, is executive secretary.

Oct. 19-24—

American Welding Society. Annual meeting and convention to be held in conjunction with the National Metal exposition, Philadelphia. Headquarters are at Bellevue-Stratford Hotel. M. M. Kelly, 29 West Thirty-ninth street, New York, is secretary.

Oca. 20-24—

American Society for Metals. Annual meeting and exposition to be held at the Benjamin Franklin hotel, Philadelphia. W. H. Eisenman, 7301 Euclid avenue, Cleveland, is secretary.

Oct. 20-25—

Dairy Industries Supply Association Inc. Exposition and meetings are to be held in Toronto, Canada. Roberts Everett, 232 Madison avenue, New York, is executive vice president.

Oct. 30-Nov. 1—

Society of Automotive Engineers. National Aircraft Production meeting and engineering display to be held at the Biltmore Hotel, Los Angeles. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary.

Nov. 12—

American Die Casting Institute. Annual meeting to be held at the Cleveland hotel, Cleveland. Kenneth C. Castleman, 420 Lexington avenue, New York, is secretary.

Business and Sales Briefs

INCREASE in its productive capacity of high-alloy steels with the installation of a new 35-ton electric melting furnace, and another still to be put into service, has been announced by Allegheny-Ludlum Steel Corp. The increased melting capacity at Brackenridge, Pa., is part of the company's expansion program to increase the company's production of special metals by approximately 50,000 tons annually.

♦

After being district manager of the Houston territory for seven years, J. Walter Snavely has returned to the sales department of the conveying and engineering products division of Chain Belt Co. at Milwaukee. Other changes include that of W. B. Marshall, formerly assistant sales manager, being appointed as sales manger of the division. B. E. Sivyer, former San Francisco branch manager has been transferred to Milwaukee as assistant sales manager of the chain belt and transmission division. Replacing Mr. Sivyer as branch manager at San Francisco is S. Y. Warner who has been connected with the company since 1930.

♦

SKF Industries Inc. have established new branch offices and warehouse at 1419 South Flower street, Los Angeles.

♦

Plans for the construction of a factory at Fort Wayne, Ind., to produce turbo-superchargers for airplanes, have been announced by General Electric Co. At the present time the company is manufacturing this part at Lynn, Mass., and a factory also is nearing completion at Everett, Mass., for similar manufacture. The latter plant is a new "blackout" factory.

♦

With offices at 4905 Santa Fe avenue, Los Angeles, Archie A. Morris will represent McKenna Metals Co., Latrobe, Pa., in southern California.

♦

In recognition of his abilities and success in former positions, C. W. Pearsall has been appointed general sales manager of Ahlberg Bearing Co., Chicago. Joining the organization in 1919, he has been successively a salesman in Chicago and Philadelphia, branch manager in both cities, and manager and distributor of sales in Chicago.

♦

Erection of a new foundry is under way at Ampco Metal Inc., Milwaukee. The new addition will be devoted largely to production work, allowing continuous handling of long runs of castings. When completed the foundries will be the largest in the country devoted primarily to the production of aluminum bronze of controlled analysis.

♦

Formerly manager of the Canadian subsidiary of Handy & Harman, James W. Colgan has been made sales manager. His headquarters will now be at 82 Fulton street, New York.

♦

A. T. Carter, Rochester, N. Y., has been appointed representative of Keystone Carbon Co. in upper New York for the company's line of motor and generator brushes and resistors. The following distributors

for Keystone's line of self-lubricating bronze bearings have also been named: The Ohio Ball Bearing Co., 6541 Euclid avenue, Cleveland; Indiana Bearings Inc., 510 North Capitol street, Indianapolis; and West Virginia Bearings, 8 Virginia street, East Charleston, W. Va.

♦

New distributors who are specialists in the machinery field have been appointed by the Drafto Co., Cochranton, Pa. These are Walter D. Briggs, 117 Liberty street, New York, and Arthur J. Moore, 1048 North Lockwood avenue, Chicago.

♦

A new warehouse at 131 Clarendon street, Boston, has been established by Cutler-Hammer Inc. as a service to the trade, and complete up-to-the-minute stock will be carried for immediate delivery. W. E. Addicks, district manager, is in charge of the Boston territory.

♦

Injection molding facilities of the plastics division, Erie Resistor Corp., Erie, Pa., have been expanded recently by the addition of three automatic presses, the largest of which is rated at 16 oz.

♦

Offices of the Dardelet Threadlock Corp., 55 Liberty Street, New York, have been moved to the Machinery building, East Grand boulevard, Detroit.

♦

Formerly with Foote Bros. Gear & Machine Corp., Chicago, Herbert J. Braun has been appointed Detroit sales representative for Kennametal by McKenna Metals Co., Latrobe, Pa. His offices are at 14425 Mark Twain avenue, Detroit, where John S. Roney is in charge.

♦

Succeeding W. A. Finn, called to active duty with the Navy, R. M. Cleveland has been appointed manager of the Boston office of Worthington Pump & Machinery Corp., Harrison, N. J. Mr. Cleveland has been with the company 24 years.

♦

Construction on a neoprene plant at Louisville, Ky., by E. I. du Pont de Nemours Inc., has been started. It is expected that production of this material will exceed the present combined total U. S. production of all synthetic rubbers including neoprene itself. The Deepwater, N. J., plant of the company has already reached 6000 tons a year, and this figure, it is believed, will be raised to 9000 by the end of the year.

♦

M. J. Tennes Jr., president of Shafer Bearing Corp., has entered active service as Captain in the United States Army Air Corps with permanent station at the Air Corps Advanced Flying Schools, Phoenix, Ariz. During the absence of Capt. Tennes, the management of the company will be under the direction of John F. Ditzell, vice president and general manager.

♦

A new two-story building is being added to the present plant of Michigan Molded Plastics Inc., Dexter, Mich.

♦

With a maximum capacity of 15 million pounds of sheet brass per month, a new brass sheet rolling mill is being built by Revere Copper & Brass Inc., in Chicago.

♦

Manufacturing facilities of Paragon Electric Co. are being moved from Manitowoc to Two Rivers,

Wis., where 25,000 square feet of floor space doubles the area of the former plant. Executive headquarters will remain in Chicago.

♦

Located at 1011-19 West Washington boulevard, Chicago, the Dayton Rubber Mfg. Co., Dayton, O., has opened an office for sales, engineering and warehousing purposes. This additional space is required because of increased demand for drives.

♦

Transfer of Howard M. Dawson of the Cleveland office of Jessop Steel Co., Washington, Pa., to the Detroit office has recently been announced by the company. D. J. Hanna is branch manager of the new Detroit warehouse and office at Woodbridge and Walker streets.

♦

Succeeding the late G. K. Simonds, Alvan C. Simonds, chairman of the board, Simonds Saw & Steel Co., Fitchburg, Mass., has been named president and general manager. G. K. Simonds Jr. is assistant general manager.

♦

H P L Mfg. Co., 2015 East Sixty-fifth street, Cleveland, has been established to specialize in stampings in small quantities, ranging from 25 to 5000 pieces in metal, fiber and other sheet materials. The company was established by Ray Hedberg, Kermit Peterson and Melvin Lorentz who have been associated with the stamping industry in Minneapolis for eleven years.

♦

New offices and show rooms have just been occupied by Hunter Electro-Copyist Inc., Syracuse, N. Y., manufacturers of photo-copying machines and sensitized papers and linens. This is the second move within three years necessitated by expansion of the company's activities. The new quarters are in the Duguid building, 428 South Warren street.

♦

Ruthman Machinery Co. has moved into its large new plant and office located at 1819 Reading road, Cincinnati. The output of its line of coolant pumps has increased 25 per cent during the first month the company has been in its new building, and a 40 per cent increase is expected during the second month, according to A. H. Ruthman.

♦

E. H. Alexander has been appointed engineer of the industrial control division of General Electric company's industrial department. In this capacity, Mr. Alexander will co-operate with the manager of this division.

♦

Previously vice president of export sales, J. Louis Reynolds has been appointed general sales manager of the Reynolds Metals Co. Inc., for the duration of the present national emergency.

♦

R. E. Spencer Geare has become associated with Thermoid Rubber division of Thermoid Co., Trenton, N. J. He will concentrate on improvement of the company's multiple-V belt and F.H.P. belt programs and expansion of V-belt sales.

♦

After an illness of several months, George L. Markland Jr. died recently at the age of 73. Mr. Markland was chairman of the board of Philadelphia Gear Works and also had been, since 1935, president of the Philadelphia Board of Trade.